VOL. V. NO. 2.

JOURNAL

OF THE

American Society of Naval Engineers.

MAY, 1893.

PUBLISHED QUARTERLY BY THE SOCIETY.

WASHINGTON, D. C.
R. BERESFORD, PRINTER, 617 E STREET, N. W.
1893.

Entered at the Postoffice at Washington, D. C., as Second Class Matter.

CONTENTS.

PAGE.

EXPERIMENTAL COMPARISON OF A SINGLE-CYLINDER ENGINE WITH A TRIPLE-EXPANSION ENGINE. By Chief Engineer Isherwood, U. S. Navy, 271

CIRCULATING PUMPS. By Assistant Engineer Henry G. Leopold, U. S. Navy, 295

EXPERIMENTS ON THE ARRANGEMENT OF THE SURFACE OF A SCREW PROPELLER. By William G. Walker, Esq., 300

MARINE BOILERS, 327

SUPPORTS FOR BORING BARS FOR STERN TUBES AND STRUTS. By Chief Engineer C. R. Roelker, U. S. Navy, 340

THE WORTHINGTON MARINE FEED WATER HEATER, 342

STEAM PIPES—THEIR MATERIAL, WORKMANSHIP AND ARRANGEMENT. By W. J. Nowers Brett, 347

THE CONTRACT TRIAL OF THE U. S. S. *BANCROFT*. By Passed Assistant Engineer Robert S. Griffin, U. S. Navy, 365

THE CUNARD STEAMER *CAMPANIA*, 385

MARINE BOILER FURNACES, 413

THE TRANSMISSION OF HEAT THROUGH TUBE PLATES. By A. J. Durston, Engineer-in-Chief of the British Navy, 436

DUTY TRIAL OF A BLAKE HIGH DUTY PUMPING ENGINE, AT THE CITY WATER WORKS, NEWTON, MASS. By F. W. Dean, M. E., 465

EFFECT OF AUXILIARIES ON ECONOMICAL SPEED. Additional Discussion by Frederick Merium Wheeler, Esq., Associate, 474

NOTES—

The Floating Derrick at Cramps', 477

Thornycroft's Screw Turbine Propeller, 479

Manganese Steel, 483

The Strength of Torpedo Boats, 483

Effect of Shell Fire on Tanks of Petroleum Residuum, 486

New Shipbuilding Programme for British Navy, 486

SHIPS, 492

MERCHANT STEAMERS, 522

BOOK REVIEWS, 527

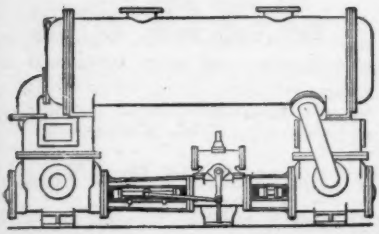
CORRECTIONS AND ADDITIONS TO THE LIST OF MEMBERS AND ASSOCIATES OF THE SOCIETY, 531

INTERNATIONAL ENGINEERING CONGRESS, 532

PAGE.
271
295
300
327
340
342
347
365
385
413
436
465
474
477
479
483
483
486
486
492
522
527
531
532

DAVIDSON
STEAM PUMPS

FOR ALL SITUATIONS.



SURFACE CONDENSERS,
With or without Air and Circulating Pumps.

VERTICAL AIR AND CIRCULATING PUMPS
Combined or Independent.

WATSON'S METALLIC PACKING,
As generally used in New Vessels of the United States Navy.

PUMPS for MARINE SERVICE
A SPECIALTY.

VERTICAL PUMPS of any size and for any pressure, from 2 inches to 9 inches diameter of water cylinder.

COMBINED AIR AND CIRCULATING PUMPS, HORIZONTAL OR VERTICAL, and from the smallest size to the requirements of 10,000 H.P. of main engine. These combined pumps obtain with the minimum of space and weight the maximum of efficiency.

I **guarantee** to deliver **as much water** with my **single cylinder** pumps as can be delivered by a duplex pump having **two cylinders** of **equal area,** and with from **20 to 30 per cent. less expenditure of steam,** and consequently of fuel, and to deliver the quantity under pressure or otherwise, with as little moment of intermission.

I also manufacture **Blowing Engines** and **Air Compressors,** direct-acting or crank and flywheel type, for limited pressure, as may be required on ship-board.

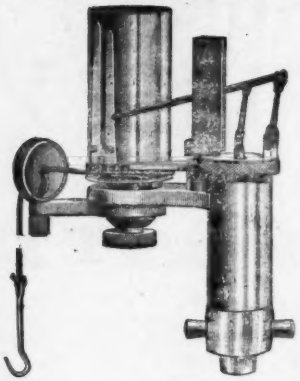
M. T. DAVIDSON.

Principal Office and Works:

43 to 53 Keap Street, Brooklyn, New York.

Branch Offices, 77 Liberty Street, N. Y. 51 Oliver Street, Boston, Mass.

THE ASHCROFT MANUFACTURING CO.,



SOLE MANUFACTURERS OF

The Tabor Steam Engine Indicator.

Approved and adopted by the U. S. Government on the new Cruisers.

ALSO STEAM AND VACUUM GAUGES, WITH PATENT SEAMLESS DRAWN TUBES.

Steam Traps, Packer Ratchet Drills, Stanwood Cutters, Marine Clocks, Engine Revolution Counters, Test Gauges, Oil Cups, Test Pumps, Locomotive Spring Balances, Steam Whistles, Salinometers, and all Instruments for Measuring Steam, Gas or Water.

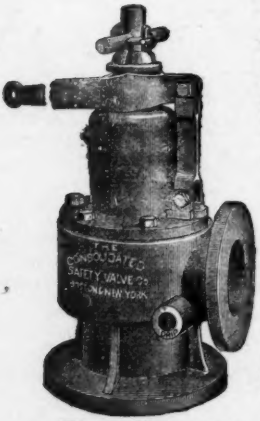
Office and Salesroom: 111 Liberty Street, New York.

Factory: Bridgeport, Conn.

THE CONSOLIDATED SAFETY VALVE CO.

SOLE MANUFACTURERS OF THE

ONLY SOLID NICKEL-SEATED SAFETY VALVE



Approved by U. S. Board Supervising Inspectors. Adopted by U. S. Navy, and furnished to all the Steel Cruisers.

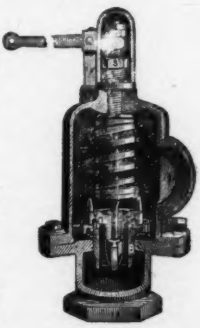
MADE SPECIALLY TO COMPLY WITH RECENT REQUIREMENTS OF U. S. STEAMBOAT INSPECTORS.

Angle of Seat 45°, and Allowed 50 Per Cent. Higher Rating Than Government Lever Valve.

The ONLY Safety Valve Made with Richardson's Patent Adjustable Screw Ring.

We are prepared to furnish the BEST SAFETY VALVE MADE, and at very low prices when QUALITY and CAPACITY are considered. Address—

FOR MARINE AND STATIONARY BOILERS.



Send for Illustrated Catalogue.

THE CONSOLIDATED SAFETY VALVE CO.,

Salesrooms: 111 Liberty Street, New York. **Works: Bridgeport, Conn.**

S. T. WELLMAN, President. C. H. WELLMAN, Superintendent.
S. H. CHAUVENET, Vice-President. RICHARD PETERS, Jr., Secretary.
JOHN P. CROZER, Treasurer.

Wellman Iron and Steel Co.,

MANUFACTURERS OF

Highest Grade
Open Hearth
Low Phosphorus

STEEL PLATE

FOR FIRE-BOX, FLANGE AND BOILER PURPOSES.

PLATES UP TO 126 INCHES IN WIDTH.

General Office and Works, Thurlow, Del. Co., Pa.

CHICAGO, ILL.:
656 THE ROOKERY.
E. W. CRAMER, Agent.

NEW YORK, N. Y.:
7 NASSAU STREET.
J. H. BELCHER, Agent.

THE COLUMBIAN IRON WORKS AND DRY DOCK CO.,

BALTIMORE, MD., *May* 1st, 1893.

VACUUM OIL COMPANY,
Rochester, N. Y.

Gentlemen: It gives me pleasure to reply to your letter, and state that we used Vacuum Marine Engine Oil and 600 W. Mineral Cylinder Oil on the preliminary and official trials of the U. S. Steamer *Detroit* with the most satisfactory results.

The *Detroit* is a 2,000-ton cruiser equipped with two triple-expansion engines of 6,000 indicated horse power; H.P. cylinder, 26½″; I.P., 39″, L.P., 63″ and stroke 26″; revolutions per minute, 180; steam pressure above atmosphere at H.P. cylinder, 165 pounds; piston speed, 780 feet per m., developing a speed of ship of 19.27 knots per hour.

The oil service was at all times during the preliminary and official trials sufficient on engines and cylinders, and at no time was there any necessity for supplementing this by hand oiling.

Examination of cylinders and the various crank pins and important journals since the trial has shown them to be in perfect condition.

Personally I would unhesitatingly use your oil on any ship in which I was interested, whatever might be the speed or the pressure per square inch on the journals, as I am confident it is equal in all respects to the best quality of olive or sperm oils.

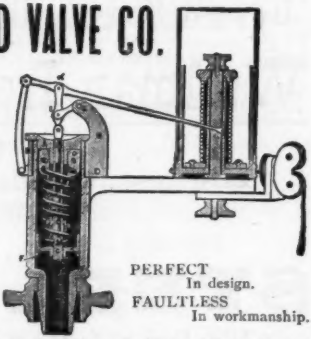
Yours very truly,

THE COLUMBIAN IRON WORKS & DRY DOCK CO.,
WILLIAM T. MALSTER, *Prest.*

CROSBY STEAM GAGE AND VALVE CO.

Sole Manufacturers of the

Crosby Steam Engine INDICATOR



Approved and adopted by the U. S. Government. It is the standard in nearly all the great Electric Light and Power Stations of the United States. It is also the standard in the principal Navies, Government Ship-Yards, and the most eminent Technical Schools of the world.

PERFECT
In design.
FAULTLESS
In workmanship.

Letter from W. D. Weaver, late Assistant Engineer U. S. Navy, designer of the elaborate Indicator Testing Apparatus at the New York Navy Yard, used for testing and calibrating the Indicators of U. S. Naval Vessels.

GENTLEMEN: In reply to your request for the results of my experience with the various makes of indicators tested at the New York Navy Yard for use on the horse-power contract trials of Naval Vessels, I am happy to state that the Crosby Indicator was not surpassed in any particular by other indicators, and was decidedly superior in workmanship and in the drum mechanism.

The test cards of your indicators were invariably satisfactory, and the same may be said of the operation of the instruments on both slow and fast-running engines.

Very respectfully, (Signed) W. D. WEAVER.

This Company is also Sole Manufacturer of **CROSBY** Pop Safety Valves and Water Relief Valves. Improved Steam Pressure Gages.

Main Office and Works, 93 and 95 Oliver Street, BOSTON, MASS.

Branches: NEW YORK. CHICAGO. LONDON, ENG.

Star Brass Manufacturing Co.

BOSTON, MASS.

Office and Factory, 33 LANCASTER STREET.

MANUFACTURERS OF

Steam, Water and Vacuum Gages,

With Non-Corrosive Movements. Specially Adapted for Marine Use.

Revolution Counters,
Marine and Locomotive Clocks,
Sight Feed Lubricators & Oil Cups,
"Pop" Safety Valves and

BOILER APPLIANCES IN GENERAL.

CATALOGUE FURNISHED ON APPLICATION.

THE HYDRO-PNEUMATIC ASH EJECTOR.

(Patented in the United States and Europe.)

Prevents soiling of the decks and annoyance of the passengers. Relieves the firemen from hoisting ashes, and gives more time to attend to fires. Does not injure paint or plating at side of vessel.

The work of discharging can be done in port as well as at sea.

The repairs will cost less than with the old methods.

Between 30 and 40 of them in successful operation on transatlantic and coastwise steamships, river steamers, yachts, etc.

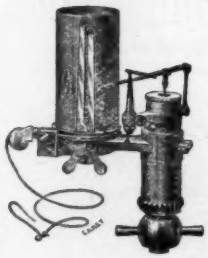
HORACE SEE, 1 Broadway, New York.

SOLE MANUFACTURER UNDER THE PATENTS.

(ORIGINAL STEAM GAUGE CO.)

AMERICAN STEAM GAUGE CO.

INCORPORATED 1854.



SOLE MANUFACTURERS OF THE

Thompson Improved Indicator.

MORE THAN 5000 IN USE.

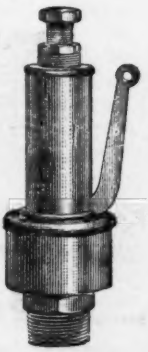
Adopted by the U. S. Navy

for use on all the New Cruisers and Gun boats to be built.

ALSO MANUFACTURERS OF

AMERICAN POP SAFETY VALVE,

the only Automatic Self-Adjusting Safety Valve ever produced for locomotive, stationary, marine and portable boilers.



ALSO MANUFACTURERS OF

BOURDON PRESSURE GAUGES

WITH LANE'S IMPROVEMENT.

Water Gauges, Gauge Cocks, Whistles, Revolution Counters, Marine Clocks, Hydrometers, Pyrometers, Salinometers, and all instruments incidental to the use of Steam.

36 CHARDON ST., BOSTON, MASS.

SEND FOR CATALOGUE.

Quintard Iron Works,

N. F. PALMER, Jr. & CO.

Manufacturers and Builders of

Engines, Boilers and Machinery

OF EVERY DESCRIPTION.

Ave. D, 11th and 12th and Dry Dock Streets, East River, New York.

Repairs done at short notice and with dispatch.

Office, 742 East 12th Street.

F. P. BUDDEN, President. W. E. VOLZ, Treas. and Gen'l M'g'r.

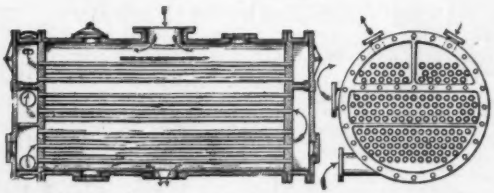
STANDARD CONDENSER CO.,

BUILDERS OF

LIGHTHALL SURFACE CONDENSERS,

BRASS AND COMPOSITION CONDENSERS.

SOLE BUILDERS.



Patent Combined Surface Condenser and Feed Water Heater.

PATENT BRASS SCREW GLANDS, TUBE HEADS, PACKINGS, &c.

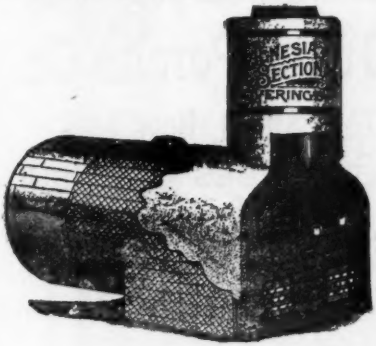
CONDENSER REPAIRS.

15 State Street, New York.

Send for Circular.

SECTIONAL COVERINGS OF THE GREAT

Magnesia Conservator.



It keeps in the **HEAT**, so saving the **STEAM** and sparing the **COAL.** Furthermore, it is itself preserved, yielding neither to heat nor to vibration.

Light, porous, attractive, economical. Thus far the acme of attainment in heat insulation.

Branch Office:
119 Franklin St., Buffalo, N. Y.

R. A. KEASBEY,
54 Warren St., New York.

SELDEN'S PATENT PACKINGS,

FOR STUFFING BOXES OF ENGINES, PUMPS, AIR COMPRESSORS, GLOBE VALVES, ETC.

THE "**SELDEN**" is in use in the U. S. Navy, and the largest Marine and Stationary plants in this and other countries, and its merits have been testified to by the repeated orders where it has once been introduced. The materials of which it is composed are entirely free from any substances which will either **score** or **corrode** rods and plungers, and will keep them tight with **less friction** than any packings on the market. A trial will convince you of the justice of these claims. It is put up in handy shape for the consumer and dealer.

Round, with Rubber Core, in sizes (varying by sixteenths) from ¼ to 2 inches diameter.
Round, with Canvas Core, in sizes (varying by sixteenths) from ¼ to 1½ inches diameter.
Square, with either Rubber or Canvas Core, in sizes (varying by sixteenths) from ⅝ to 2 inches diameter.

RANDOLPH BRANDT,
38 CORTLANDT STREET, NEW YORK.

HIGH INSULATION is just as essential in electric wiring as are strong pipes in plumbing and in steam engineering.

The Bishop Gutta Percha Co.,

420-426 East 25th Street, New York,

Makes the HIGHEST GRADES OF INSULATED WIRES AND CABLES FOR UNDERGROUND, UNDER-WATER AND AERIAL USE, and also for INSIDE WIRING. Remember that in insulation WATER-PROOF means FIRE-PROOF.

TRY OUR FLEXIBLE CORDS.
SEND FOR NEW ILLUSTRATED CATALOGUE.

HENRY A. REED, Secretary and Manager.

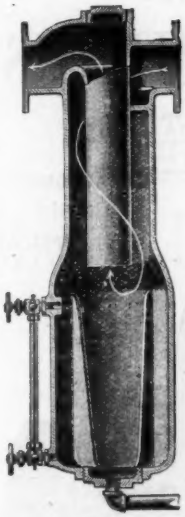
GRAPHITE . . . LUBRICATION.

Prof. R. H. Thurston, in a series of tests, demonstrated that pure flake graphite largely increased the lubricating value of all greases and oils.

Graphite being the softest and smoothest of all known minerals, and a most excellent conductor of heat, it is readily seen why all authorities on lubrication agree that pure flake graphite, properly prepared, is by odds the best natural lubricant known to science.

FOR PAMPHLET AND FOR INFORMATION concerning the many uses of that wonderful mineral, GRAPHITE, address

JOS. DIXON CRUCIBLE CO., JERSEY CITY, N. J.



THE

STRATTON SEPARATOR.

This is the only apparatus that automatically separates water from steam, and secures the maximum of Economy, Efficiency and Safety.

Specified by the Navy Department, and used on all the New Cruisers of the

UNITED STATES NAVY,

and by the Leading

Steamship Lines, Railroads, Water Works and Electric Light Companies.

THE GOUBERT MFG. CO.,

(Sole Manufacturers),

32 Cortlandt Street. NEW YORK.

STRAIGHT LINE CENTRIFUGAL SEPARATOR

AND GREASE EXTRACTOR.

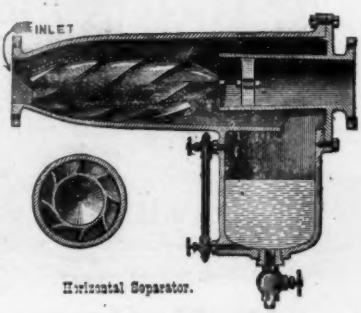
... INSURES ...

DRY STEAM — PURIFIED EXHAUST.

The entrained water in the steam, by being whirled along the shell and lodged in a separate chamber or receptacle, is positively isolated from the flow of the *Dry Steam*, thus avoiding all possibility of its being picked up again or producing condensation by contact.

In use on the gunboats of **The United States Navy**, and on the steamers of the Fall River Line, Providence Line, Pacific Mail Steamship Co., New York & Cuba Steamship Co., R. & O. Navigation Co., etc., and in all the principal central stations of electric lighting companies.

Vertical Separator.



Horizontal Separator.

The Spiral Channels on Cone COMPEL a Positive Centrifugal Motion Giving an Outward Impulse to the Heavier Particles in the Steam.

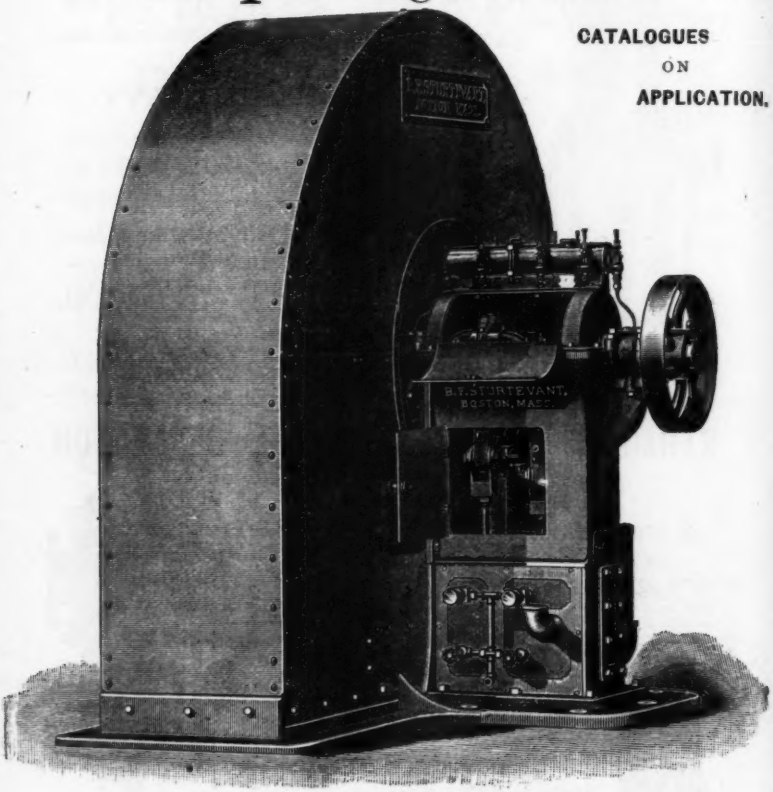
SEND FOR DESCRIPTIVE CIRCULAR.

JOS. DE RYCKE, SOLE MAN'F'R,

145 Broadway, New York.

THE STURTEVANT
STEAM BLOWERS
FOR FORCED COMBUSTION.

CATALOGUES
ON
APPLICATION.



B. F. STURTEVANT CO.,
BOSTON, MASS.

91 Liberty St., New York. 16 S. Canal St., Chicago. 135 N. Third St., Phila.

W. & A. FLETCHER CO.,

North River Iron Works,

MARINE ENGINES, BOILERS, ETC.

Hudson, 12th and 14th Streets,

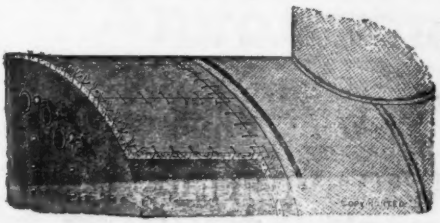
HOBOKEN, N. J.

TAKE FERRY FROM FOOT OF WEST 14TH ST., N. Y.

ASBESTOS FIRE-FELT

FOR COVERING

MARINE BOILERS.



SECTIONAL,
DURABLE,
REMOVABLE,
FIRE-PROOF.
WATER-PROOF.

Approved by U. S. NAVY DEPARTMENT and in use on the NEW CRUISERS of the U. S. NAVY.

H. W. JOHNS MANUFACTURING CO.,

Sole Manufacturers of H. W. Johns' Asbestos Roofing, Sheathing, Building Felt, Asbestos Steam Packings, Boiler Coverings, Roof Paints, Fire-Proof Paints, etc.

VULCABESTON Moulded Piston-Rod Packing Rings, Gaskets, Sheet Packing, etc.

Jersey City, Chicago, Philadelphia, 87 MAIDEN LANE, New York. Boston, Atlanta, London.

Established 1858.

RIVETS.

STEEL OR IRON.
STRAIGHT OR SWELL NECK.
MADE IN SOLID DIES.

WRITE FOR PRICES.

MICHIGAN BOLT AND NUT WORKS,

DETROIT, MICH.

ORFORD COPPER CO.

37 Wall Street, New York.

ROBERT M. THOMPSON, Prest.

COPPER INGOTS, WIRE BARS AND CAKES

FERRO NICKEL

AND

FERRO NICKEL OXIDE

FOR USE IN PREPARING

NICKEL STEEL FOR ARMOR PLATES.

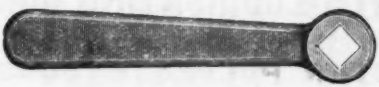
.. THE ..

BILLINGS & SPENCER CO.

MANUFACTURERS OF

OF EVERY DESCRIPTION. **WRENCHES** OF EVERY DESCRIPTION.

MACHINIST'S TOOLS

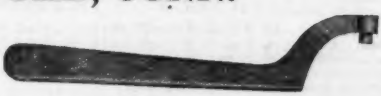


AND **DROP FORGINGS**

Of Copper, Bronze, Iron and Steel.

The Billings & Spencer Co.,

HARTFORD, CONN.



D. W. PEDRICK. H. C. AYER.

PEDRICK & AYER,

MANUFACTURERS OF

UNIVERSAL AND PLAIN

MILLING MACHINES.

Richard's Patent Open-side Planing and Shaping Machines.
Special Tools for Railway Repair Shops.
Portable and Stationary Cylinder Boring Machines.
Universal and Centrifugal Grinding Machines.

OFFICE AND WORKS:

1001-3 Hamilton St., 1002-4 Buttonwood St.,

PHILADELPHIA, PA.

THE PULVER LUBRICATING COMPOUND

—MANUFACTURED BY—

PETER PULVER & SONS,

NO. 214 FRANKLIN STREET, - - - NEW YORK.

No Freezing of Oil! Works Absolutely Without Drip or Slop.

Particularly adapted for all classes of Marine and Stationary Machinery.

An experience of 25 years brought to bear in the compounding of the above article, producing a strictly first-class compound of unsurpassed quality, perfect in action, and of the greatest economical value, that is guaranteed to give entire satisfaction in all climates and under all conditions. Is not a paraffine grease, but composed of the highest grade animal oils and greases, non-gumming waxes and other substances necessary to produce the proper cooling and wearing body essential in a first-class lubricant. Catalogues containing full information mailed on application.

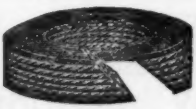
Selling agents for the celebrated RAINBOW FLANGE AND JOINT PACKING. The greatest article of its kind ever invented. Retains its elasticity under any conditions. Is not affected by steam, heat, ammonia or alkalies. Will not blow out under the highest pressure of steam. Information and samples furnished. Send for same.

THE ECLIPSE BOILER GASKET.—A newly-invented and patented novelty in this line that will prove a revelation. Any one can make all sizes of hand and manhole gaskets out of a box of stock material in a few minutes that will give better results than the old style. No waste whatever. The Rainbow Stock used in its construction. Send for a descriptive pamphlet of this wonderful invention.

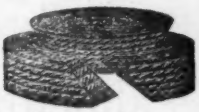
NOTICE.—On February 25, 1884, the firm of Cook & Pulver was dissolved by mutual agreement. All their trade-marks were assigned to Mr. Adam Cook, and the right to continue the manufacture of the products was reserved by Mr. Peter Pulver. The firm of PETER PULVER & SONS will continue the manufacture of the product known as Albany Lubricating Compound, under the name of Pulver Lubricating Compound.

GARLOCK PACKINGS

Patented.



ELASTIC RING.



SECTIONAL RING.

SPIRAL PACKING.

FOR STEAM, WATER AND AMMONIA.

A Perfect Fit to the Rod and Box. Subjected to a Perfect Lubricating Process.

Correspondence solicited, and catalogues sent on application.

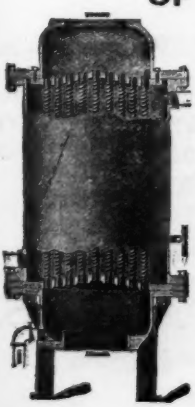
THE GARLOCK PACKING CO.

194 WATER STREET, NEW YORK.

PALMYRA, N. Y. ROME, GA. OMAHA, NEB. PHILADELPHIA, 24 N. 4th St.
PITTSBURGH, 99 First Ave. CHICAGO, 94 Franklin St.

THE WAINWRIGHT MFG. CO.

OF MASSACHUSETTS.



SOLE MANUFACTURERS OF

CORRUGATED COPPER TUBE

Feed Water Heaters

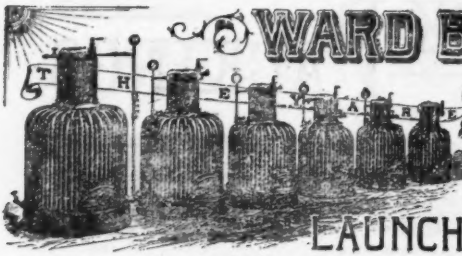
Surface Condensers

—AND—

Expansion Joints.

No. 8 OLIVER ST., - BOSTON

Send for Catalogue "N."



UNEQUALLED FOR EFFICIENCY.

"Superior to any other Boiler offered for test."

B. F. TRACY, Secretary of the Navy.

"A standard which future Boilers must equal or surpass."

GEO. W. MELVILLE, Engineer-in-Chief, U. S. N.

"Admirably designed on sound principles."

B. F. ISHERWOOD, Chief Engineer U. S. N., President.

"In accessibility, ease of making repairs, simplicity and interchangeability of parts, the Ward Boiler has considerable advantage."

CHARLES H. LORING, Chief Engineer U. S. N., President.

THE U. S. COAST DEFENSE VESSEL, *MONTEREY*, AND FIFTY OTHER BOATS OF THE WAR AND NAVY DEPARTMENTS AND REVENUE MARINE, ARE FITTED WITH THESE BOILERS.

ADDRESS:

WARD BOILER WORKS,

CHARLESTON, W. VA.

THE ATLANTIC WORKS,

INCORPORATED 1853.

Border, Maverick and New Sts., East Boston, Mass.

BUILDERS OF

STEAMSHIPS, TOW BOATS, AND STEAM YACHTS,

In Steel, Iron and Wood.

Marine Engines, Marine and Stationary Boilers, Tanks and

GENERAL MACHINERY.

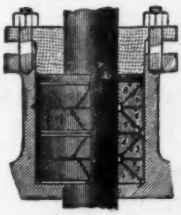
BAIRD'S

FRESH WATER EVAPORATORS
FRESH WATER DISTILLERS
AUTOMATIC BOILER FEEDERS
AUTOMATIC STEAM TRAPS

77 Liberty Street, New York. **51 Oliver Street, Boston.**

49 Keap Street, Brooklyn.

KATZENSTEIN'S Self-Acting Metal Packing,



For PISTON RODS, VALVE STEMS, etc., of every description, for Steam Engines, Pumps, etc. etc.

Adopted and in use by the principal Iron Works and Steamship Companies, within the last twelve years, in this and foreign countries.

FLEXIBLE TUBULAR METALLIC PACKING, for slip-joints on Steam Pipes, and for Hydraulic Pressure; also METAL GASKETS for all kinds of flanges and joints.

For full particulars and reference, address,

L. KATZENSTEIN & CO., Machinists,
357 West Street, NEW YORK.

Also, Agents for the McColl-Cumming Patent "Liquid Rudder Brake" and Cumming "Double-Acting, Water-Tight Bulkhead Door."

THE GEO. F. BLAKE M'F'G CO.,

BUILDERS OF EVERY VARIETY OF

Pumping Machinery

95 & 97 Liberty Street,

NEW YORK.

185 Devonshire St.,

BOSTON.

535 Arch Street,

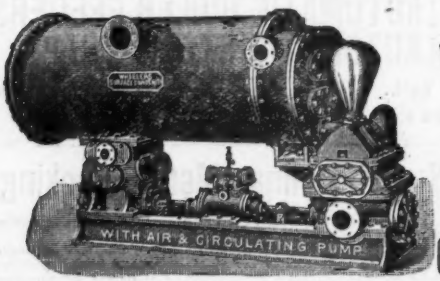
PHILADELPHIA.

STEAM PUMPS FOR NAVAL USE A SPECIALTY.

Send for Illustrated Catalogue.

WHEELER CONDENSER & ENGINEERING CO.

Nos. 39 and 41 CORTLANDT STREET,



Proprietors and Manufacturers of

WHEELER'S

IMPROVED

PATENT SURFACE CONDENSER.

☞ **Also, the Wheeler-Admiralty Condenser, with Patent Screw Glands.**

Light Weight Surface Condensers for Steam Launches a Specialty.

N. B.—The following U. S. War Vessels are furnished with the Wheeler Admiralty Condenser, combined with Air and Circulating Pumps: "New York," "Dolphin," "Vesuvius," "Montgomery," "Detriot," "Indiana," "Massachusetts," and the Triple Screw Cruisers 12 and 13.

THE COWLES ENGINEERING COMPANY,

FOOT 44th ST., BROOKLYN, N. Y.

Engineers and Shipbuilders.

The Cowles Water Tube Boiler.

High Class Marine Engines.

Triple and Quadruple Expansion Engines for

Electric Light and Trolley Work.

STEEL AND COMPOSITE VESSELS OF ANY DESCRIPTION UP TO 250 FT. IN LENGTH.

Special Designs. High Class Work. Correspondence Solicited.

MORGAN IRON WORKS

Foot East Ninth Street, New York.

STEAM MACHINERY AND BOILERS,

FLOATING STEAM DERRICK.

AMPLE DOCK ROOM

THE

ARMINGTON & SIMS

ENGINE CO.

PROVIDENCE, R. I.,
U. S. A.

We have furnished over 3000 Engines for Electric Light and Power Stations in all parts of the world.

WE BUILD

SPECIAL ENGINES

Vertical and Horizontal,

DIRECT ACTING

COMPOUND and TRIPLE COMPOUND.

HOISTING ENGINES.

DECK HOISTS

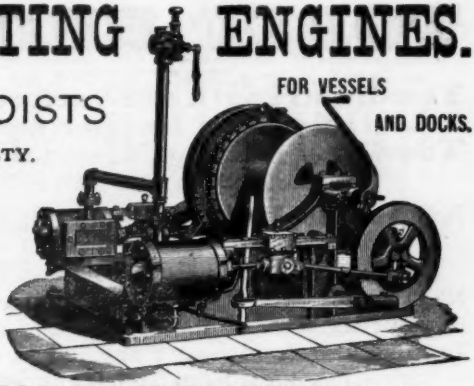
A SPECIALTY.

ENGINES AND BOILERS

COMBINED FOR

Dock Hoisters

FOR VESSELS AND DOCKS.



LIDGERWOOD MANUFACTURING CO.,

96 Liberty Street, New York.

34 and 36 West Monroe Street, Chicago.
197 to 203 Congress Street, Boston.
99 First Avenue, Pittsburgh.
5 and 7 N. First St., Portland, Oregon.
610 N Fourth Street, St. Louis.
505 Main Street, Louisville, Ky.
15 North Seventh Street, Philadelphia.

Sales Agents: Fraser & Chalmers, Salt Lake City, Utah; Helena, Montana.
Hendrie & Bolthoff Mfg. Co., Denver, Col.

BUCYRUS STEAM SHOVEL AND DREDGE CO.

BUCYRUS, OHIO,

(Will soon remove to South Milwaukee, Wis.)

STEAM DITCHING MACHINES.

MAKERS OF **STEAM DREDGES** FOR

Engineers, Contract Work, Harbor and River Use, Canals, Slips, &c.

TOBIN BRONZE.

Tensile strength one-inch rods upwards of 79,000 lbs. per sq. inch.
Torsional strength equal to the best Machinery Steel.

NON-CORROSIVE IN SEA WATER.

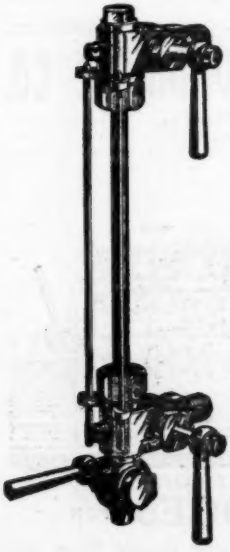
Round, Square and Hexagon Bars for bolt forgings, &c., Pump Piston Rods, Yacht Shafting, Ship Sheathing, Spring Wire, Rolled Sheets and Plates for Pump Linings, and Condenser Tube Sheets, etc.

CAN BE FORGED AT CHERRY RED HEAT.

ANSONIA BRASS AND COPPER CO.,

SOLE MANUFACTURERS,

Send for Circular. **19 and 21 Cliff Street, NEW YORK.**



STEAM SPECIALTIES.

ASBESTOS PACKED

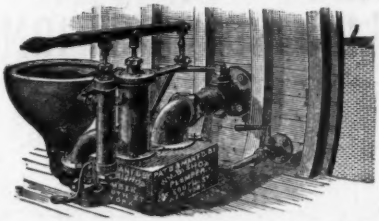
WATER GAUGES
AND
GAUGE COCKS.

THE ABOVE STYLES ARE NOW IN USE ON MANY OF THE GOVERNMENT CRUISERS, AND ARE RECOGNIZED AS STANDARD BY THE ENGLISH NAVY.

THE FAIRBANKS CO., NEW YORK.

PATENT PUMP WATER CLOSET

For Yachts, Pilot Boats, Naval Ships, Etc.



In Use on U. S. SHIPS

MIANTONOMOH,
CINCINNATI,
MARBLEHEAD,
AND OTHERS. *RALEIGH.*

In Use on ALL KINDS OF
YACHTS AND LAUNCHES
From 30 Feet up to 285 Feet.

FOR ABOVE OR BELOW WATER LINE.
SUPPLY IS TAKEN DIRECT FROM THE SEA AND DISCHARGED INTO SEA BELOW WATER LINE.

WILLIAM BISHOP,

Plumber, Steam Fitter and Coppersmith,

205 SOUTH ST., NEW YORK.

TELEPHONE 4811 CORTLANDT. TELEPHONE 4811 CORTLANDT.

CORNELL, HISCOX & UNDERHILL,

Wrought and Cast-Iron Pipe,

FITTINGS, VALVES AND COCKS.

STEAM SPECIALTIES.

SUPERIOR QUALITY

IRON AND STEEL BOILER TUBES,

TO CONFORM TO UNITED STATES NAVY SPECIFICATIONS AND TESTS.

13 GOLD STREET,

NEW YORK.

McINNES'

ENGLISH

PATENT METALLIC COMPOSITION

AND PROTECTIVE COATING

FOR BOTTOMS OF STEEL AND IRON SHIPS.

In use by all the Principal Lines of European and American Steamers and Steam Yachts. It is the "Fastest" Coating for Ships or Yachts.

GEORGE N. GARDINER, Sole Agent for the U. S.

Telephone Call, 3070 Cortlandt.

Agency in San Francisco,
THE CALIFORNIA PAINT CO.,
22 Jessie Street.

53 SOUTH ST., N. Y.

LONG-DISTANCE TELEPHONE, "2473 CORTLANDT."
CABLE ADDRESS, "GALAPAGOET."

J. FRIEDENSTEIN,

COPPER

In All Grades and Shapes,

METALS

Of Every Description.

MINERS' AND SMELTERS' AGENT

For the Sale of

COPPER ORES AND PRODUCTS,

213 Pearl Street, New York.

R. M. GILMOUR M'F'G CO.,

MANUFACTURERS OF

ASBESTOS and FELTING Materials.

FIBROID Fire-Proof Sectional Covering.

Approved by the U. S. Navy Department for use on the new cruisers of the U. S. Navy.

Magneso-felt and Combination Insulators,

For Boilers, Steam Pipes and all Heated Surfaces. Made specially to comply with requirements of U. S. Steamboat Inspectors.

STANDARD HAIR FELT.

MINERAL WOOL OF ALL GRADES—SLAG AND ROCK.

THE TRADE SUPPLIED.

82 JOHN STREET, - - NEW YORK.

ESTABLISHED 1857.

AMERICAN SHIP WINDLASS CO.

P. O. Box 53, PROVIDENCE, R. I.

BUILDERS OF

Windlasses and Capstans

For all Classes of Vessels.

NAVAL WINDLASSES,

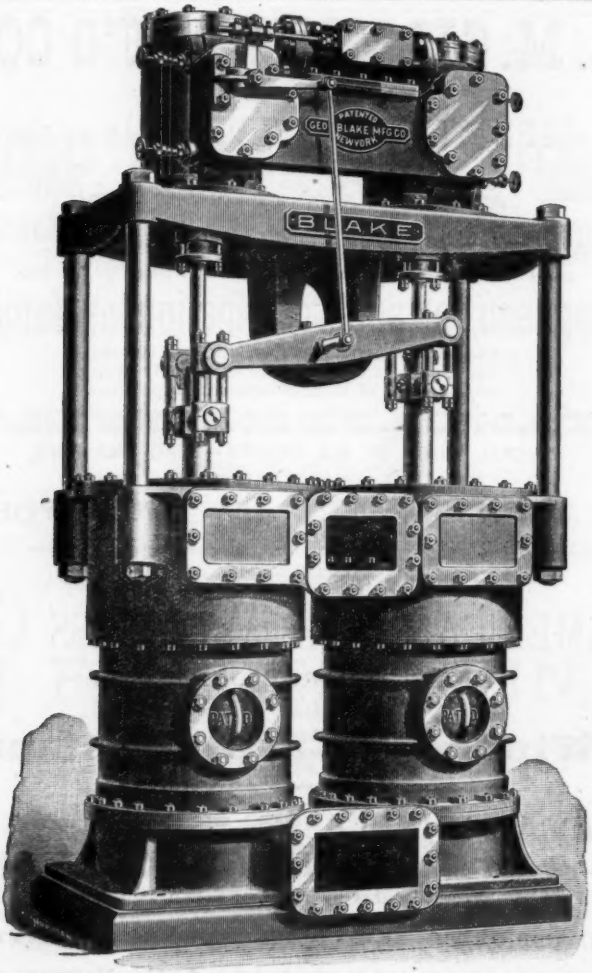
NAVAL CAPSTANS,

Worked either by Hand or Steam Power.

IN USE ON THE NEW CRUISERS OF THE U. S. NAVY.

This Company is prepared to execute orders for Ship Machinery of all kinds in either Iron or Steel, with promptness and despatch.
Send for Illustrated Catalogue. Address

FRANK S. MANTON, Agent.



VERTICAL TWIN-CYL. AIR PUMP of U. S. Cruiser "NEW YORK."

(BLAKE SYSTEM.)

Also furnished for the U. S. Cruiser "Columbia" and Battleships "Massachusetts" and "Indiana."

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. V. MAY, 1893. No. 2.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published)
Chief Engineer H. WEBSTER, U. S. Navy.
Passed Ass't Eng'r F. H. BAILEY, U. S. N. Passed Ass't Eng'r IRA N. HOLLIS, U. S. N.
Passed Ass't Eng'r W. M. MCFARLAND, U. S. N. Ass't Eng'r W. W. WHITE, U. S. N.

EXPERIMENTAL COMPARISON OF A SINGLE-CYLINDER ENGINE WITH A TRIPLE-EXPANSION ENGINE.

BY CHIEF ENGINEER ISHERWOOD, U. S. NAVY.

There is held yearly in France a meeting or congress of the chief engineers of the various associations, in that country, of the proprietors of steam apparatus for the insurance of boilers, and for the improvement of the management of the same; incidentally, these associations are connected with the experimental examination of all kinds and details of steam machinery, and the investigation of the causes affecting its economic performance. Their chief engineers are men of the highest scientific and practical attainments, most of them having a wide-spread celebrity; they print annually for private distribution a *compte-rendu* of their sittings, and from the *annuaire* of their 14th Congress held at Marseilles on the 4th, 5th, 6th and 7th of May, 1890, the writer has taken the data of the experiments hereinafter de-

scribed. These experiments were made with the greatest care, and the various measurements involved can be accepted without reserve. Only the observed facts have been taken; the quantities calculated from them, and the deductions made not being found in the original reports.

The experiments in question were selected by the writer because they enabled him to make a comparison between the economic performances of simple or, better called, single-cylinder engines and triple-expansion engines, better called three-cylinder engines, under conditions which, though unfavorable in every respect for the single-cylinder engine comparably with the triple-expansion one, were yet more nearly what they should be than can ordinarily be had.

The single-cylinder condensing engine has generally been worked with steam of low pressure used with small measures of expansion in unjacketed cylinders, and the economic results obtained have been compared with those given by triple-expansion engines, steam jacketed and using very high-pressure steam with large measures of expansion. This great discrepancy in the conditions prevents any proper comparison of the two kinds of engines as types; in fact it simply shows the economic advantages of high-pressure steam used with proportionally large measures of expansion over low-pressure steam with proportionally small measures of expansion.

As regards the particular two engines experimentally in question, the conditions compare as follows, taking in each case the trials producing the highest economic result:

Space displacement of the piston of the single-cylinder engine per stroke, 2.901405 cubic feet; and of the piston of the large cylinder of the triple-expansion engine, 7.573269 cubic feet.

Initial steam pressure in pounds per square inch above zero on the piston of the single-cylinder engine, 100.025; and on the piston of the small cylinder of the triple-expansion engine, 149.805 pounds.

Measure of expansion with which the steam was used in the single-cylinder engine, 5.916 times; and in the triple-expansion engine, 15.359 times.

Number of double strokes of pistons made per minute in the case of the single-cylinder engine, 88.066666; and in the case of the triple-expansion engine, 106.6.

Indicated pressure on the piston of the single-cylinder engine 25.06344 pounds per square inch; and equivalent indicated pressure on the piston of the large cylinder of the triple-expansion engine 25.6 pounds per square inch.

Indicated horses power developed by the single-cylinder engine 67.746278; and by the triple-expansion engine 177.970905.

There was about the same back pressure in the condensers of both engines; and the cylinders of both were well steam jacketed on top, bottom and sides, all the jackets being filled with steam of nearly the boiler pressure.

The steam jacketing should be more efficient in the case of the triple-expansion engine with its greatly increased cylinder surface in contact with the boiler steam, and with its greater measure of expansion, which made a greater difference of pressure for the jacket temperature to act on.

All the above cited conditions were very largely in favor of the triple-expansion engine; as they all largely influence the economy of the steam in its production of power by means of a steam engine, nevertheless, the indicated horse power was obtained with the single-cylinder engine for an hourly expenditure of 16.453398 pounds of water, or, more properly, of 18,115.56 Fahrenheit units of heat, while the indicated horse power was obtained with the triple-expansion engine for an hourly expenditure of 15.540164 pounds of water, or, more properly, of 16,991.68 Fahrenheit units of heat, the difference being 6.204 per centum of the cost of the power in units of heat with the single-cylinder engine, or 6.614 per centum of the cost of the power with the triple expansion engine, in favor of the latter engine.

There can be no doubt that this difference in favor of the triple-expansion engine of about one-sixteenth of the fuel would have been reversed had the conditions been equal for both engines, and that the single-cylinder engine *efficiently steam jacketed*, would have exceeded the triple-expansion engine in the economic production of power by even a higher proportion, had it worked under the same conditions.

If the premise be admitted that the cylinder of a single-cylinder engine can be so efficiently steam jacketed as to prevent all steam liquefaction in the cylinder, then, other things being the same, the single-cylinder engine must exceed both the triple-expansion and the compound or two-cylinder engine, in economy, by the losses of effective pressure (not of total pressure) experienced in transferring the steam from one cylinder to the other in those types of engine. The above premise also includes that steam jacketing will prevent steam liquefaction in the cylinders of the triple-expansion and compound engines.

Whether the premise expresses a fact or not, can be determined by experiment only, and, unfortunately, the reports of the experiments in question are totally incomplete in data that could be used for such a purpose. Had mean indicator diagrams been furnished in the two cases, or had the steam pressures been given at the point of cutting off the steam, at the end of the expansion, at the commencement of the compression, also the mean pressure of the expanding steam alone after the closing of the cut-off valve and the mean back pressure against the pistons, both inclusive and exclusive of the cushioned pressure, the fact could have been ascertained for the experiments in question. A very little labor would have accomplished this when the diagrams were before the experimenters, and would have added immensely to the value of the experiments. Of equal importance would have been a repetition of the experiments with the steam omitted from the jackets. A vast proportion of the experiments which have been made in steam engineering are, like these, so incomplete, that they do not satisfactorily settle anything, and they as often mislead as guide. There seems to be no comprehension that the performances of steam engines depend almost wholly on the limitations and conditions employed, and not simply on type of engine, and that the same engine will give as many results as there are changes in the limitations and conditions under which it is used.

When an experiment is made, all the limitations and conditions which affect it should be ascertained and accurately stated,

together with an exact description of the machine, including the dimensions of all its principal parts.

Notwithstanding that in the case of the triple-expansion engine with its much higher initial steam pressure, and the enormously greater measure of expansion with which the steam was used, and the greater area of the jacket surfaces in proportion to weight of steam consumed in equal time, and the greater difference between the jacket temperature and the temperature due to the mean pressure of the expanding steam in the cylinders, the per centum of the steam evaporated in the boiler, liquefied in the jackets, was greater in the single cylinder engine. With the triple-expansion engine, the liquefaction in the steam jackets was 7.676 per centum of the total steam evaporated in the boiler, while with the single-cylinder engine that per centum was 12.579, or over one-half more, a very important result if it can be shown by other and sufficiently numerous experiments to be general.

The interesting fact that, in the case of the single-cylinder engine, decreasing the measure of expansion with which the steam was used from 5.916 to 4.765 times, other things remaining the same, so little affected the economy of the performance, that the slight difference may be properly attributed to the small errors almost unavoidable in such experiments. Even with the highest steam pressures used in practice, and with steam-jacketed cylinders, there is great doubt whether the maximum economic effect obtainable from expansion requires the expansion to be carried beyond eight times.

The simultaneous determination of the indicated horses power, of the horses power required to work the engine *per se* or unloaded, and of the horses power developed by the engine at a friction brake, allows the horses power expended in overcoming the friction of the load, or what is the same thing, the coefficient of friction, to be ascertained, and for the case of a working steam engine. The experiments with the single-cylinder engine gave the indicated horses power and the horses power developed at the friction brake, and from them the coefficient of friction may be closely inferred, although the horses power required to work the unloaded engine were unfortunately not ascertained.

When the exactness of the experiments and the importance and rarity of their determinations are considered, their value will become manifest. Suspicions regarding the accuracy of the results from the single-cylinder engine may be entertained by some persons, owing to the small cost per hour in feed water and in units of heat of the horse power developed, but there is no reason to doubt them. Other experiments, unexceptionally made on other single-cylinder engines *efficiently steam jacketed*, have given even greater economies.

In what, then, does the superiority of the compound and of the triple-expansion engine over the single-cylinder engine consist? Obviously, in their superior practical adaptation for the use of high pressures. They make high pressure as practicable, so far as the engine is concerned, as the low pressure that preceded their introduction, and they do it by enormously lessening the *maximum effective* pressure of the steam upon the piston without lessening the *mean effective* pressure, other things remaining the same. As they make higher initial pressure practicable, so, also, do they make greater measures of expansion both practicable and economical, the measure of expansion up to a certain point carrying economic gain with it bearing some ratio to the initial pressure. It would be found quite impracticable to use the large cylinders required for the development of large powers with high-pressure steam considerably expanded in single-cylinder engines; the maximum strain on the fastenings, and on the crank pin and crank-shaft journals, necessary to obtain the resulting indicator or mean pressure, would make the employment of such a type of engine out of the question. The compound engine can enormously reduce this maximum strain, preserving the same indicator pressure, and the triple-expansion engine can proportionally still more reduce it. And there must be remembered that the thickness of metal in all parts of steam engines must be proportioned to the maximum strains—not to the mean strains—so that as regards weight of engine alone for a given power, the difference, conditions equal, is not as great in favor of the single-cylinder engine as at first may appear.

Further, with each additional cylinder there go additional crank pin and main journals, so that the maximum pressure upon these journals is not only greatly lessened, but the number of journals working under this lessened pressure is greatly increased. In other words, for a given power developed in the single-cylinder engine and in the triple-expansion engine, there are three times the length of crank pin and main journals to receive a considerably less maximum pressure with the triple-expansion engine than with the single-cylinder engine. The friction of these journals is thus much lessened, and the danger of "heating" obviated, matters of prime importance in practical steam engineering.

The superiority of the multiple-cylinder engines is, then, mechanical rather than economic, steam of the same regimen being used with steam jacketing or steam superheating in both cases.

In the following pages will be found, in as much detail as the writer could obtain, the description of the engines of the two types experimented with, the description of the manner of making the experiments, and the tables containing the data and results of the experiments.

EXPERIMENTS MADE TO DETERMINE THE ECONOMIC EFFICIENCY OF DUPLICATE STEAM-JACKETED, CONDENSING, SINGLE-CYLINDER ENGINES.

The following experiments were made by Mr. L. Bour, the engineer director of the Lyonnaise Association of the Proprietors of Steam Apparatus, France, in conjunction with Mr. Dougados, the mining engineer at the Rive-de-Giers, to determine the economic efficiency of the duplicate steam-jacketed, condensing single cylinder engines, constructed by Messieurs L. Piguet for the public hospital of the city of Lyons, France. The experiments were required for the purpose of ascertaining whether the guarantees of the contractors had been fulfilled, and whether the city officials were warranted in accepting the engines.

The engines were duplicates, with horizontal cylinders coupled directly to the same shaft, and, for distinction, are called "right hand" and "left hand." They were experimented with successively, the first on the 23d of March, 1890, the last on the suc-

ceeding 30th March. Both experiments were made in precisely the same manner, with practically the same boiler pressure and piston speed, but with considerable difference in the measure of expansion with which the steam was used. In the "right hand" engine the steam was cut off when the piston had completed 11.71 per centum of its stroke, and in the "left hand" engine the cutting off was done when the piston had completed 16.05 per centum of its stroke, the experiments thus determining the difference of economic efficiency due to such considerable difference of cut off, while the initial steam pressure on the piston and the speed of the piston remained sensibly constant.

The cylinders were steam jacketed on the sides and on both ends, the jackets being filled with steam taken directly from the boiler by special pipes. The following are the principal dimensions of the cylinders:

Number of cylinders for each engine	1.
Diameter of the cylinder	15.748316 inches.
Net area of the piston, exclusive of half the area of the piston rod	192.945287 square inches.
Stroke of the piston	2.62472 feet.
Displacement of the piston per stroke	2.901405 cubic feet.
Diameter of the piston rod	2.165393 inches.

The power developed by each engine was measured simultaneously by the indicator and by a Prony friction brake applied to the fly wheel situated on the main shaft and between the two cylinders, the work done during the experiments being the overcoming of the friction of this brake at uniform speed; consequently, the power developed in each experiment was absolutely uniform, and, as the boiler pressure and the regimen of the steam were maintained unchanged during each experiment, the indicator diagrams were nearly duplicates of each other.

Two indicators were employed, one at one end, the other at the other end of the cylinders with which they were connected by short pipes of large diameter. A diagram was taken every half hour from each end of each cylinder, the results for each experiment, in the table hereafter given, being the mean from nineteen sets of diagrams. The springs of the indicators were

afterwards carefully corrected at the temperatures with which they were used.

The number of double strokes made by the pistons was taken by a counter. The different gauges and thermometers used were corrected from standards.

The feed water was carefully weighed into a tank from which it was pumped into the boiler. The water level was maintained constant during each experiment, and was left at exactly the same height at the end as at the beginning of the trials.

The water of liquefaction drawn off from the jackets was carefully and continuously received in closed vessels so as to avoid any loss by vaporization under the atmospheric pressure. This water was accurately weighed, and its temperature in the vessels was 312.8 degrees Fahrenheit, or about 20 degrees less than the boiler temperature, part of which was due to external refrigeration, but the larger part was probably due to the fact that, neglecting the water of liquefaction in the jackets due purely to external radiation, the steam liquefied in the jacket is liquefied by and upon the exterior surfaces of the cylinder, and the resulting water has, consequently, the temperature of those surfaces, which is necessarily much lower than the boiler temperature; in fact, the temperature of this water is the mean temperature of those surfaces, an observation now made for the first time. The less the cross area of the jacket pipes, other things equal, the greater will be the difference of pressure between the jackets and the boiler for the delivery of equal weights of steam in equal time, and to the insufficient area of these pipes may doubtless be ascribed no small part of the discrepancies found in experiments with different steam-jacketed engines using steam with substantially the same regimen, as regards the fraction of the total weight of steam evaporated in the boiler, liquefied in the steam jackets.

In all experiments with steam jackets, a gauge should be attached to them to show the pressure within, for without such evidence there can be no certainty that a sufficiently close approximation to the boiler pressure is present in them, and the difference may be very great, so great indeed as to prevent the jackets from producing any economic effect. The writer has

never known of an experiment in which the pressure in the jackets was ascertained, though obviously it should always be instrumentally determined, as much so, in fact, as the pressure in the boiler or in the condenser. The temperature of the water of liquefaction leaving the jackets should also be experimentally ascertained; it cannot be assumed to be the same as the boiler temperature, even when the pressure in the jacket is the same as the pressure in the boiler.

When the steam jackets are cast with the cylinder in one piece, there is much difficulty in cleaning out the cores in order that the metal of the cylinder may come in contact with the jacket steam, and not be prevented by an intervening layer of non-heat-conducting baked clay. And when "liners" are used, as one end of the liner has to be free in order to move in the outside casting without fracture, when the liner and it have different temperatures, which they necessarily always have, there is always correspondingly more or less possibility of steam leakage during half the time from the jackets into the cylinder, and thence into the exhaust without doing work on the piston. When this is the case very little water of liquefaction will be obtained from the jackets, as it will revaporize under the lessened pressure of the exhaust, the resulting steam passing into the condenser. There is no certainty of non-leakage in this respect, unless the free end of the "liner" passes through a regular stuffing box, the elastic packing of which is compressed by a follower set up with screw bolts. Is this ever done? The ordinary method is to make the free end of the liner merely a metallic fit when cold. Such a joint is never steam tight, and the thickness of a fine hair multiplied by the circumference of the cylinder will produce an area sufficiently great to easily leak steam enough to neutralize the economy of the jacket. There is no use to make expensive and troublesome steam jackets unless sufficient additional expense is incurred to secure them steam tight. The same remarks apply to endeavoring to make a steam-tight fit between the end of the "liner" and the cylinder cover. It cannot be permanently accomplished owing to the differences of expansion in the different parts, subjected as they are to great differences

of temperature; consequently, the "liner" steam jackets in practical use are more or less sham jackets, and, as they leak more or less, give experimentally a correspondingly great variety of economic results from a considerable gain down to none at all. The whole uncertainty, as regards the economy of steam jacketing has thus arisen, bad design and ignorant experimenting have combined to produce doubts on a subject which admits of none; for it is physically impossible to envelope the cylinder of a working steam engine, be its type or size what it may, by a temperature greater than the highest temperature within, and largely greater than the mean temperature within, without producing a well-marked economic gain due to the prevention of steam liquefaction inside the cylinder and the revaporization of the water of that liquefaction principally during the exhaust, involving thereby great loss of heat. Whenever a steam jacket shows but little or no economic gain, and particularly when the water of liquefaction drawn from it is a very small fraction of the boiler vaporization, there is certainty that one or more of the causes above pointed out is in action and producing the effect. The jacket, according to circumstances, is either not supplied with steam of the boiler pressure, or the cores have not been wholly removed, or the "liners" leak.

The writer is here speaking of steam jackets which envelope the entire cylinders—not merely a portion of their barrels.

Singularly enough, experimenters carefully test their indicators and compare their thermometers with a standard in order to avoid small errors, but they never ascertain whether the engine which is the subject of the experiment is in good working order, or whether its pistons, valves and jackets leak steam at every junction.

The first duty of an experimenter is to ascertain whether his appartus is tight and in working order, and then to experimentally ascertain—inferring nothing—by count, weight, pressure or measure, every datum involved in the final determination. All this is rarely done, and the failure in these respects is the real cause of the discordant and contradictory results often obtained in steam engineering. With competent experimenters and

well made apparatus, they would quickly vanish. There are no miracles in nature. The same causes will always produce exactly the same effects, and if they seem not to do so in any particular case, it is because their action has been modified or changed by the unsuspected interference of other causes, which, if intelligently sought for, will be found.

The steam pipes, cylinders, etc., were well protected by non-heat-conducting material, and the observations were taken by two observers who mutually checked each other, and were never absent from the engine room during the continuance of the trials.

The friction brake functioned perfectly during both experiments. It was lubricated with a solution of soap in water, and the following practical facts are given in connection therewith: *New* soap-water should be continuously used; soap-water once used should never be used again. One hundred and fifty pounds of soap should be dissolved in one hundred pounds of water, the water being a little warm in order to secure the complete solution of the soap.

The simultaneous determination of the indicated power, and of the power developed by the engine at the friction brake, enables a point of interest, not well understood, to be elucidated, namely, to what causes and in what proportions, is the difference between the indicated pressure and the pressure on the piston corresponding to the power at the brake, to be ascribed.

The difference between the indicated horses power and the horses power developed by the engine at the brake, overcomes two distinct resistances governed by different laws, though both are resistances due to friction. The first is the power required to overcome the resistance of the working parts *per se*, of the engine, such as the friction of all the pistons and piston rods, of the slide valves, of the guides of the crosshead, of the crank pin and crank shaft and other journals, etc., etc., of the *unloaded* engine, that is, of the engine when the work is thrown off and the unloaded steam pistons make the same number of double strokes in a given time as before the work was thrown off. This power will be in the direct ratio of the number of double strokes

made by the steam pistons in a given time, as the friction pressure is constant for all speeds; it can be ascertained experimentally by simply disconnecting the load and taking an indicator diagram at any speed of piston, the resulting pressure being the pressure for all speeds. When this diagram is taken, the cut-off gear must be thrown out and the steam used without expansion, or there will be an error in the mean pressure, which will be given too great, owing to the inertia and momentum of the indicator piston and attachments.

The second is the power required to overcome the friction of the load, and is directly proportional to the load, as the coefficient of friction is constant; that is to say, this power is proportional to what remains of the indicated horses power after deduction of the power required to work the unloaded engine.

The pressure on the piston required to work the unloaded engine, was not experimentally ascertained, and has, therefore, been assumed at one pound per square inch, which deducted from the indicator pressure of 25.06344 pounds per square inch during the experiment of March 23d, leaves 24.06344 pounds per square inch for the net pressure. Now the pressure on the piston corresponding to the horses power developed by the engine at the brake was 22.58524 pounds per square inch, subtracting which from the 24.06344 pounds, leaves 1.47820 pounds per square inch for the friction of the load, which pressure is 6.143 per centum of the net pressure 24.06344 pounds.

Calculating in the same manner for the experiment of March 30th, with its indicated pressure of 29.4199 pounds per square inch of piston, its net pressure of 28.4199 pounds, and its pressure of 26.159646 pounds corresponding to the horses power developed by the engine at the brake, there results the pressure 2.260254 pounds per square inch for overcoming the friction of the load, which pressure is 7.953 per centum of the net pressure 28.4199 pounds.

The mean of the two determinations is

$$\left(\frac{6.143 + 7.953}{2} =\right) 7.048$$

per centum of the net pressure. The writer has always adopted

the co-efficient 7.5 per centum of the net pressure, which, indeed, is Morin's co-efficient resulting from his numerous experiments on friction, as the practically correct one for the conditions of ordinary practice, and it has been confirmed by many expriments on steam engines.

As the pressure required to work the engine *per se* or unloaded is constant, there should be, with continuously increasing indicated pressure, a continuously diminishing ratio between those pressures and the pressures on the piston corresponding to the power developed by the engine at the friction brake, and such has been found to be the case in the many experiments made by the French engineers up to a certain point beyond which the ratio, on the contrary, increased with increasing pressures. As the cause of this reversal has not been explained, the writer may here state that this result arises from the fact that after a certain indicated pressure is reached, producing a certain pressure on each square inch of the journals, the oil between the working surfaces begins to be forced out, thus increasing their friction, and as the pressure to overcome increasing friction must increase proportionally, the co-efficient of the friction of the load increases, thus producing the observed effect.

The experiments determine the difference under the experimental conditions of the economic efficiency of the steam when used with an expansion of 5.916 times, and when used with an expansion of 4.765 times, which include the effect of the waste space at the ends of the cylinders between the piston when at the end of its stroke and the face of the cut-off valve. This space, in the absence of direct measurement, has been assumed at one-sixteenth or 0.0625 of the space displacement of the piston per stroke.

The experimental conditions were as proper ones for the purpose as can be obtained. The initial pressure of the steam in the cylinder was, in both cases, 100 pounds per square inch above zero; the cylinders were completely steam jacketed with steam of about 106.5 pounds pressure per square inch above zero, having a temperature of about 332 degrees Fahrenheit. The number of double strokes made by the pistons was nearly the same,

averaging 87.5 per minute. The back pressure against the pistons was 3.48 pounds per square inch above zero with the large expansion, and 4.00 pounds with the small expansion, the difference being too small to sensibly affect the result. The mean total pressure on the piston was 28.54344 pounds per square inch above zero for the large expansion, having the mean temperature 247 degrees Fahrenheit, the difference between which and that of the boiler steam presumably in the cylinder jackets was 85 degrees. The mean total pressure on the piston, with the small expansion, was 33.42 pounds per square inch above zero, having the mean temperature 256 degrees Fahrenheit, the difference between which and that of the boiler steam is 76 degrees. The difference between the temperature of the feed-water in the two cases was insignificant, but is included in the calculation for the number of Fahrenheit units of heat respectively consumed per hour. This calculation assumes the boiler to have been fed with the whole of the water of liquefaction drawn from the steam jackets, and, for the residue, with water drawn from the condenser, each at its appropriate temperature. The proper horses power to be compared with these units of heat, are the total horses power, as the work of overcoming the back pressure is as much work performed by the expanding steam as the work reprepresented by the indicated horses power.

Under these circumstances, the total horse power was developed with a consumption of 15,906.92 Fahrenheit units of heat per hour in the case of the large expansion, and with a consumption of 16,117.30 Fahrenheit units of heat per hour in the case of the small expansion, the difference being 1.2915 per centum of the latter quantity, and 1.3226 per centum of the former quantity, per centages easily falling within the limits of error of such trials, and warranting the inference that the economy of the steam was the same with both measures of expansion.

The data and results of the experiments will be found in the following table, the quantities of which are so fully described in it that no further explanation is needed:

TABLE CONTAINING THE DATA AND RESULTS OF THE TRIALS MADE BY MR. L. BOUR, THE ENGINEER-DIRECTOR OF THE LYONNAISE ASSOCIATION OF THE PROPRIETORS OF STEAM APPARATUS, FRANCE, AND BY MR. DOUGADOS, THE MINING ENGINEER AT THE RIVE-DE-GIERS, TO DETERMINE THE ECONOMIC EFFICIENCY OF THE DUPLICATE SINGLE-CYLINDER, STEAM-JACKETED CONDENSING ENGINES CONSTRUCTED FOR THE PUBLIC HOSPITAL OF LYONS, FRANCE, BY MESSRS. L. PIGUET AND COMPANY OF THAT CITY.

	Right hand Engine.	*Left hand Engine.*
Date of the experiment	23d March, 1890.	30th March, 1890.
Duration of the experiment in consecutive hours	10.	10.
Pounds of feed water pumped into boiler from the tank, per hour	9,744.426	11,656.494
Pounds of feed water pumped into boiler from the steam jackets of the cylinders, per hour	1,402.139	1,390.455
Total pounds of feed water pumped into boiler per hour	11,146.565	13,046 949
Fahrenheit units of heat put into the total pounds of feed water pumped into the boiler per hour	12,272,615.47	14,385 816.97
Pounds weight of steam entering the cylinders per hour	9,744.426	11,656.494
Fahrenheit units of heat put into the steam entering the cylinders per hour	10,728,829.39	12,852,674.53
Per centum which the water from the steam jackets is of the total feed water pumped into boiler	12.579	10.657
Pressure in the boiler in pounds per square inch above the atmosphere	92.135	91 922
Pressure in the boiler in pounds per square inch above zero	106.823	106.610
Initial pressure in the cylinder in pounds per square inch above zero	100.025	100.025
Back pressure against the piston in pounds per square inch above zero	3 480	4.0001
Fraction completed of the stroke of the piston when the steam was cut off	11.71	16.05
Number of times the steam was expanded	5.916	4.765
Indicated pressure on the piston, in pounds per square inch	25.06344	29.4199
Net pressure on the piston in pounds per square inch	24.06344	28.4199
Total pressure on the piston in pounds per square inch	28.54344	33.4200

	Right hand Engine.	*Left hand Engine.*
Back pressure in the condenser in pounds per square inch above zero	1.720	2.250
Pressure on the piston in pounds per square inch corresponding to the horses power developed at the friction brake	22.58524	26.159646
Number of double strokes made per minute by the piston	88.066666	87.016666
Mean speed of the pistons in feet per minute	462.300683	456.788771
Temperature of the feed water in degrees Fahrenheit, drawn from the condenser	82.4	86.0
Temperature of the boiler steam in degrees Fahrenheit	332 4	332.25
Indicated horses power developed by the engine	67.746278	78 573635
Net horses power developed by the engine	65.043286	75.902870
Total horses power developed by the engine	77.152690	89.256962
Horses power developed by the engine at the friction brake	61.047734	69 866262
Pounds of water consumed per hour per indicated horse power	16.453398	16.604741
Pounds of water consumed per hour per net horse power	17.137150	17.189006
Pounds of water consumed per hour per total horse power	14.447409	14.617290
Pounds of water consumed per hour per horse power developed at the friction brake	18.258769	18.674176
Fahrenheit units of heat consumed per hour per indicated horse power	18,115.56	18,308.71
Fahrenheit units of heat consumed per hour per net horse power	18,868.38	18,952.93
Fahrenheit units of heat consumed per hour per total horse power	15,906.92	16,117.30
Fahrenheit units of heat consumed per hour per horse power developed at the friction brake	20,103.31	20,590.50
Pounds of water supplying the steam that entered the cylinder consumed per hour per indicated horse power	14.383707	14.835121
Pounds of water supplying the steam that entered the cylinder consumed per hour per net horse power	14.981448	15.357119
Pounds of water supplying the steam that entered the cylinder consumed per hour per total horse power	12.630053	13.059479

	Right hand Engine.	*Left hand Engine.*
Pounds of water supplying the steam that entered the cylinder consumed per hour per horse power developed at the friction brake	15 961978	16.684009
Fahrenheit units of heat put into the water supplying the steam that entered the cylinder consumed per hour per indicated horse power	15,836.78	16,357.50
Fahrenheit units of heat put into the water supplying the steam that entered the cylinder consumed per hour per net horse power	16,494.91	16,933.05
Fahrenheit units of heat put into the water supplying the steam that entered the cylinder consumed per hour per total horse power	13,905.97	14,399.63
Fahrenheit units of heat put into the water supplying the steam that entered the cylinder consumed per hour per horse power developed at the friction brake	17,574.50	18,396.11

EXPERIMENTS MADE TO DETERMINE THE ECONOMIC EFFICIENCY OF A STEAM-JACKETED CONDENSING TRIPLE-EXPANSION STEAM ENGINE.

The following experiments were made on a triple-expansion steam engine by Mr. L. Olivier, the director-engineer of the Eastern Association of Proprietors of Steam Apparatus, France, to determine whether its economic performance was in accordance with the contract guarantees. The principal results were communicated by him to the fourteenth Congress of the Chief Engineers of the different Associations of Proprietors of Steam Apparatus held at Marseilles, France, on the 4th, 5th, 6th and 7th of May, 1890, and printed in the *compte-rendu* of their sittings.

The engine was of the vertical, jet-condensing, triple-expansion type, with its three cylinders placed side by side, and having their axes in the vertical plane passing through the axis of the crank shaft beneath. Each cylinder operated directly upon the crank shaft through its own pair of cranks, and the cranks of each cylinder were so disposed that they made angles of 120 degrees with the cranks of the other cylinders.

All the cylinders were steam jacketed on top, bottom and sides. The jackets were supplied with steam by a special pipe, and their water of liquefaction was drawn off by a special pump which delivered it into the boiler.

A "separator" was placed in the steam pipe as near as practicable to the valve chest of the small cylinder, and the water of liquefaction collected in it was pumped into the boiler by a special pump, which pump also pumped into the boiler the water of liquefaction in the valve chest of the small cylinder, drawn off to the pump by a special pipe.

The waters of liquefaction from the steam jackets, from the "separator" and from the valve chest of the small cylinder, are all included in the weight of water producing the steam which developed the power of the engine during the experiments.

The steam valves of all the cylinders were three-ported slides. Only the small cylinder had an independent cut-off valve, and it was placed upon the back of the steam valve; the point of cutting off was regulated automatically by a Denis regulator and compensator. The cutting off in the intermediate cylinder, and in the large cylinder, was done at a fixed point (0.6 of the stroke of their pistons) by lap on their steam valves.

The following are the dimensions and proportions of the steam cylinders, namely:

Number of small cylinders..... 1.
Diameter of the small cylinder..... 10.630 inches.
Net area of the piston of the small cylinder, exclusive of half the cross area of its piston rod..... 85.765 square inches.
Space displacement of the piston of the small cylinder per stroke..... 1.270146642 cubic feet.
Number of intermediate cylinders..... 1.
Diameter of the intermediate cylinder..... 16.142 inches.
Net area of the piston of the intermediate cylinder, exclusive of half the cross area of its piston rod..... 201.664327 square inches.
Space displacement of the piston of the intermediate cylinder per stroke..... 2.986571 cubic feet.
Number of large cylinders..... 1.
Diameter of the large cylinder..... 25.591 inches.
Net area of the piston of the large cylinder, exclusive of half the cross area of its piston rod..... 511.375128 square inches.
Space displacement of the piston of the large cylinder per stroke.. 7.573269 cubic feet.

Diameter of the piston rods of all the cylinders..... 2.756 inches.
Stroke of the pistons of all the cylinders..... 2.1325845 feet.
Ratio of the net area of the piston of the intermediate cylinder to the net area of the piston of the small cylinder, the latter being taken as unity..... 2.351360
Ratio of the net area of the piston of the large cylinder to the net area of the piston of the small cylinder, the latter being taken as unity..... 5.962515
Ratio of the net area of the piston of the large cylinder to the net area of the piston of the intermediate cylinder, the latter being taken as unity..... 2.535774

The cylinders, valve chests, steam pipes, etc., were all thoroughly covered with non-heat-conducting material.

Two experiments were made, the last in direct continuation of the first and on the same day. One was made with the engine when doing its ordinary work, and the other when it was worked to its maximum. The purpose was to ascertain the indicated horses power developed by the engine in the two cases, and the cost of the same in pounds weight of steam consumed per hour.

The speed of the engine did not sensibly vary during each experiment, the work done by it being absolutely constant; neither, for the same reason, did the point of cutting off the steam vary.

The indicator diagrams were taken by two indicators to each cylinder, one at each end and communicating with the cylinder by a very short pipe. The diagrams were taken every half hour and were absolutely identical. The springs of the indicators were tested after the trials at the temperatures corresponding with those used during the trials, and the pressures from the diagrams were corrected accordingly.

The vacuum in the condenser is not given, nor the back pressures against the pistons, but the temperature of the water in the condenser is stated at 106 degrees Fahrenheit.

All the water pumped into the boiler was previously measured with the greatest accuracy in a tank. The water level in the boiler was maintained constant, and made exactly the same at the beginning and at the end of each trial.

The weight of steam liquefied in the steam jackets of the three cylinders was ascertained by three experiments, one made previous to the trials of the engine, another made during those trials, and the last made afterwards. The weight (212.305 pounds per hour)

given in the table, is the mean of the three, all of which were made under the conditions of the trials as regards steam pressure, point of cutting off, and piston speed. There was no perceptible difference between the quantities obtained during the two trials. The water of liquefaction from the steam jackets was pumped into the boiler during the experiments; its temperature was not ascertained, but has been taken in the following calculations to have been the same as that of the boiler; any error arising from this assumption would be too small to sensibly affect the result.

The weight of water obtained per hour from the "separator," was, in both trials, 23.942187 pounds; and the weight of water drawn from the valve chest of the small cylinder, was, in both trials, 24.339019 pounds; total, 48.281206 pounds, which may be considered the weight of steam liquefied per hour by the radiation of heat from the external surfaces of the steam pipe, "separator," and valve chest of the small cylinder, together with their connections. The temperature of these waters was not ascertained, but has been taken to be the same as the boiler temperature.

The weight of feed water, 2,505.111 pounds, pumped into the boiler per hour from the tank during the first experiment, had the temperature 106 degrees Fahrenheit, and was vaporized in the boiler under the pressure 154.214 pounds per square inch above zero, consequently each pound of this water had imparted to it (1,223.8428 — 106.0984 =) 1,117.7444 Fahrenheit units of heat, making a total of (2,505.111 × 1,117.7444 =) 2,800,073.79 units per hour.

The waters of liquefaction drawn from the steam jacket, from the "separator" and from the valve chest of the small cylinder, 260.586 pounds per hour in the aggregate, may be taken to have the boiler temperature 360.45 degrees Fahrenheit as a superior limit; they received per pound (1,223.8428 — 364.4796 =) 859.363 Fahrenheit units of heat making a total of (260.586 × 859.363 =) 223,937.97 units per hour.

Consequently, the aggregate number of Fahrenheit units of heat consumed per hour in the production of the power during

the first experiment was (2,800,073.79 + 223,937.97 =) 3,024,-011.76.

The drawing off of the waters of liquefaction from the steam pipe and valve chest was a favorable condition for the economy of the triple-expansion engine not shared by the single-cylinder engine with which it was compared. Further, the assumption that the temperatures of these waters were higher than they really were made the cost of the horse power in the case of the triple-expansion engine in Fahrenheit units of heat consumed per hour proportionally less than it really was; consequently, an allowance for the economic effect due to these two conditions should be made in favor of the economy of the single-cylinder engine, correspondingly lessening the small difference found in the previous comparison in favor of the triple-expansion engine.

Making the calculation in the same manner for the second experiment, the aggregate number of Fahrenheit units of heat consumed per hour in the production of the power during the second experiment was 3,646,862.86.

The data and results of the two trials will be found in the following table:

TABLE CONTAINING THE DATA AND RESULTS OF THE TWO TRIALS MADE BY MR. L. OLIVIER ON A STEAM-JACKETED TRIPLE-EXPANSION CONDENSING ENGINE TO DETERMINE ITS ECONOMIC EFFICIENCY. DATE OF TRIALS, MAY 2, 1889.

	1st trial, engine doing its ordinary work.	2d trial, engine developing its maximum power.
Duration of trial in consecutive hours and minutes	9.06	2.22
Pounds of feed water pumped into boiler from tank per hour	2,505.111	3,058.692
Pounds of feed water pumped into boiler from jackets per hour	212.305	212.305
Pounds of feed water pumped into boiler from "separator" and valve chest per hour	48 281	48.281
Total pounds of feed water pumped into boiler per hour	2,765.697	3.319.278
Fahrenheit units of heat put in the total pounds of feed water pumped into boiler per hour	3,024,011.76	3,646,862.86
Pounds weight of steam entering the small cylinder per hour	2,553.392	3,106.973

	1st trial, engine doing its ordinary work.	2d trial, engine developing its maximum power.
Fahrenheit units of heat put in the steam entering the small cylinder per hour	2,791,877.57	3,413,605.13
Per centum which the water from the steam jackets is of the total water pumped into boiler	7.676	6.396
Per centum which the water from separator and valve chest is of the total water pumped into boiler	1.746	1.455
Pressure in the boiler in pounds per square inch above the atmosphere	139.526	149.482
Pressure in the boiler in pounds per square inch above zero	154.214	164.170
Initial pressure in the small cylinder in pounds pounds per square inch above zero	149.805	159.761
Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off	0.35	0.35
Fraction completed of the stroke of the piston of the intermediate cylinder when the steam was cut off	0.60	0.60
Fraction completed of the stroke of the piston of the large cylinder when the steam was cut off	0.60	0.60
Number of times the steam was expanded in the small cylinder	2.576	2.576
Number of times the steam was expanded in the engine	15.359	15.359
Indicated pressure on the piston of the small cylinder in pounds per square inch	41.175	44.660
Net pressure on the piston of the small cylinder in pounds per square inch	38.175	41.660
Indicated pressure on the piston of the intermediate cylinder in pounds per square inch	24.108	28.161
Net pressure on the piston of the intermediate cylinder in pounds per square inch	22.832	26.885
Indicated pressure on the piston of the large cylinder in pounds per square inch	8.847	11.065
Net pressure on the piston of the large cylinder in pounds per square inch	8.344	10.562
Equivalent indicated pressure on the piston of the large cylinder in pounds per square inch	25.260	29.661
Equivalent net pressure on the piston of the large cylinder in pounds per square inch	23.751	28.152
Number of double strokes made per minute by the steam pistons	106.6	105 5

	1st trial, engine doing its ordinary work.	2d trial, engine developing its maximum power.
Mean speed of the pistons in feet per minute...	454.66702	449.97533
Indicated horses power developed by the small cylinder	48.654521	52.228021
Net horses power developed by the small cylinder	45.109565	48.719645
Indicated horses power developed by the intermediate cylinder	66.983798	77.437606
Net horses power develoved by the intermediate cylinder	63.438447	73.928839
Indicated horses power developed by the large cylinder	62.332586	77.155303
Net horses power developed by the large cylinder	58.788639	73.647927
Aggregate indicated horses power developed by the engine	177.970905	206.820930
Aggregate net horses power developed by the engine	167.336651	196.296411
Pounds of water consumed per hour per indicated horse power developed by the engine..	15.540164	16.049043
Pounds of water consumed per hour per net horse power developed by the engine	16.527742	16.909520
Fahrenheit units of heat consumed per hour per indicated horse power developed by the engine	16,991.68	17,632.95
Fahrenheit units of heat consumed per hour per net horse power developed by the engine..	18,071 43	18,578.35
Pounds of water supplying the steam that entered the small cylinder, consumed per hour per indicated horse power developed by the engine	14.347250	15.022527
Pounds of water supplying the steam that entered the small cylinder, consumed per hour per net horse power developed by the engine..	15.259012	15.827967
Fahrenheit units of heat put into the water supplying the steam that entered the small cylinder, consumed per hour per indicated horse power developed by the engine	15,687.27	16,505.12
Fahrenheit units of heat put into the water supplying the steam that entered the small cylinder, consumed per hour per net horse power developed by the engine	16,684.19	17,390.05

CIRCULATING PUMPS.

By Assistant Engineer Henry G. Leopold, U. S. Navy.

The circulating pump, as the name implies, has simply to circulate water through the condenser tubes. The only head it has to overcome is that produced by the friction of the water, eddies, sudden changes of velocity, and that necessary to give the water the required velocity of discharge through the condenser to supply sufficient water to the cooling surface.

This total head varies from two to twenty feet.

When no other considerations are taken into account in the choice of a pump, the advantages are decidedly in favor of the centrifugal pump.

This pump, having no valves to offer resistance and having a continuous flow of the water, is more economical for low heads than the direct acting or reciprocating pump in which the velocity of the water through the suction pipe is entirely destroyed at the end of each stroke and accelerated again at the beginning of the next stroke.

The loss of energy due to this cause will be twice as much in the direct acting pump as in the centrifugal pump. The resistance through the valve seats, and valves is another loss entirely avoided in the centrifugal pump. On the other hand these losses bear a smaller proportion to the total head when the head against which the pump has to work is great, and the high speed of the centrifugal pump for great heads would give the advantage to the reciprocating pump for high heads, as the friction of the water in the centrifugal pump would vary as the square of the speed, while in the reciprocating pump it would be the same for high as for low heads.

The use of the centrifugal pump also avoids the danger of a high pressure in the condenser heads, due to the tubes being

choked or the out board delivery being closed, and when this danger is avoided the condenser heads and water pipes may be made lighter.

The reciprocating pump is well understood and would seem to require no special consideration as a circulating pump.

The centrifugal pump on the contrary, appears to be very generally misunderstood, some regarding it as a wheel provided with vanes which rotate the water and finally throw it with sufficient violence into the discharge pipe for it to find its way to the overflow.

The centrifugal pump is, however, a pressure pump, the same as any other pump, and the rotary velocity given to the water, while it is necessary, is for generating the centrifugal pressure as soon as possible before the water finally enters the discharge nozzle.

The first point to be determined in designing a centrifugal pump is the surface velocity of the wheel necessary to produce the requisite head.

Referring to different authorities for the velocity, we find quite a variety of formulas, some giving a value more than double that of others.

In Rankine's Applied Mechanics, page 597, we find the formulas

$$h_1 = \frac{v_0^2}{g} \text{ or } v_0^2 = gh_1,$$

where

v_0 = surface velocity of the wheel

h_1 is the total head required to be overcome.

The formula for the surface velocity given by Foley in Mechanical Engineers' Reference Book is

$$h = \frac{v^2}{2g}\left(1 - \frac{r_0^2}{r_1^2}\right).$$

This, it will be observed, is less than one-half that given by Rankine.

We have here two eminent authors from whom to choose, with no assurance in following either that, on the one hand, we may not have a pump entirely too large, or, on the other, too

small. After puzzling over the formulas and analyzing the action of the wheel on the water, it will become evident that, as Rankine puts it, the centrifugal pump consists of a forced vortex within the wheel and a free vortex without the wheel. The head due to the centrifugal pressure of the water on leaving the tips of the vanes will be, as given by Foley,

$$\frac{v^2}{2g}\left(1-\frac{r_0^2}{r_1^2}\right).$$

Besides this pressure, the water has a tangential velocity, v, and the head due to it will be $\frac{v^2}{2g}$; the energy due to the velocity thus being greater than that due to the centrifugal pressure. It is desirable that this velocity should be converted into pressure before the water enters the discharge nozzle, otherwise the energy due to this velocity will be wasted in useless eddies, and the formula given by Foley will be found to apply, so that about one-half of the energy of the rotating water will be wasted in useless eddies and churning action.

In order that Rankine's formula should be applicable, it is necessary to utilize this velocity and convert it into pressure before it is delivered to the discharge nozzle.

As stated by Rankine, a free vortex is formed beyond the tips of the vanes, and the nature of such a vortex enables us to convert the velocity into pressure. The velocity of the water in a free vortex varies inversely as the distance from the center, and as the kinetic energy of this velocity disappears potential energy of pressure replaces it. It will be seen, therefore, that it is only necessary to extend the casing well beyond the tips of the vanes and allow a free vortex to be formed.

The formula for the total head, as then developed, will be

$$h=\frac{v^2}{2g}\left(1-\frac{r_0^2}{r_1^2}\right)+\frac{v^2}{2g}\left(1-\frac{r_1^2}{r_2^2}\right).$$

When the casing is not extended beyond the tips of the vanes the second part of this head becomes zero, as there would be no space for a free vortex to be formed.

This will explain why the efficiency of the centrifugal pump is often only 35 to 40 per cent., and varies all the way from this

to 80 per cent. when well proportioned. The centrifugal pump, being an inward-flow turbine reversed, it would appear that the efficiency should be, under favorable conditions, equal to that of the turbine. In the turbine we always find a small wheel in a large casing. In this case, however, guide vanes are necessary in the outer casing in order to form a vortex.

It is common in designing a centrifugal pump to contract the casing as much as possible, the tips of the vanes extending to the center of the discharge nozzle. Such a design must be inefficient, and in reality it would be a saving in weight to design the pump differently.

By increasing the efficiency from (say) 40 per cent. to 80 per cent., with a pump requiring 75 H.P. to drive it, there would be a saving of about 150 pounds of coal per hour, or, in thirty days steaming, of 108,000 pounds or about 48 tons. To effect this saving would not require an increase of 1 per cent. of what would be saved in coal and bunker space.

Another point that is often lost sight of in designing these pumps is the longitudinal thrust of the wheel, especially in wheels not encased, where the side of the disc on the opposite side from the vanes is exposed to the full discharge pressure.

In a pump with a discharge pressure of 30 pounds per square inch on a disc 24 inches diameter, the total amount will be 13,560 pounds. The pressure on the opposite side of the disc will be about one-half of this pressure. Adding to this the momentum of the water in the suction pipe it will leave about (say) 6,000 pounds end thrust, which if taken on a thrust bearing will consume considerable work. It may be readily balanced, however, by extending the vanes behind the disc, so as to rotate the water there, and keep the water in the discharge out of the pump. Holes may then be cut through the disc near the hub, the pressure equalized, and the end thrust of the wheel reduced to little or nothing.

With an encased wheel this balance would be perfect were it not for the momentum of the water entering the pump, a very slight difference of diameter, however, of the rings separating the discharge casing from the wheel will effect a perfect balance.

When the suction enters the pump on both sides, the wheel is balanced without any special provision.

The encased wheel has the disadvantage of having a joint on each side of the wheel which should be water tight, otherwise water will leak back into the body of the wheel and be circulated from the discharge to the suction side of the pump, wasting considerable energy.

The next point to consider is the shape of vane. All of the formulas given for the required surface velocity of the wheel are based upon a straight vane, where the tangential velocity of the water and the wheel are identical. When the vane is the Archimedean spiral commonly used, the formulas given apply by simply substituting the velocity of the water for that of the wheel. The ratio that one bears to the other depends only on the radial velocity through the wheel and the degree of the backward curvature, and is constant for all points within the wheel.

When any other form of vane is used, the formulas do not strictly apply.

The method of determining the head for the irregular shaped vanes was fully illustrated in an article previously published in the JOURNAL, (Vol. 3, pp. 335, 483.)

In general it may be stated that the greater the head against which the pump is designed to work the nearer the vanes should approach the straight vane, and the smaller the head the more the vanes should be inclined backward.

For circulating pumps as used on board ship, having to work only against a head of from 10 to 20 feet, the the Archimedean spiral will probably be found to be the best form of vane to use.

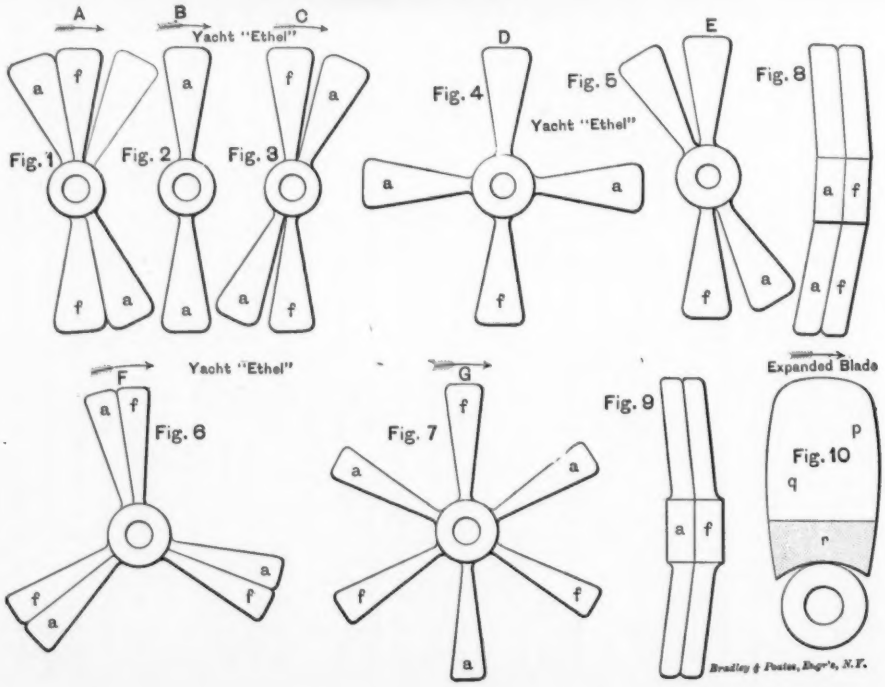
EXPERIMENTS ON THE ARRANGEMENT OF THE SURFACE OF A SCREW PROPELLER.

By William G. Walker, Esq.

[Paper read before the Institution of Mechanical Engineers, October, 1892.]

The experiments about to be described were projected and carried out by the author during 1891 in making some trials with a screw propeller invented by the late Mr. B. Dickinson, of Messrs. Navin & Co., London; and to his assistance he is especially indebted for enabling them to be undertaken. The primary object of the experiments was to ascertain the efficiency of screw propellers having the same diameter, pitch and area, but different numbers of blades, and with the blades in various positions on the boss; in other words, of screw propellers differing from one another only in the arrangement of their virtual surface.

Screws.—Seven screws were tried, which are lettered A to G, all 38½ inches in diameter with bosses 7¾ inches in diameter. They may be divided into two series: the first comprising the five screws A, B, C, D, E, and the second the two remaining screws F, G. In the five screws A to E the pitch was 64¼ inches, and the aggregate developed area of the blades was 395 square inches; A had two blades, and B, C, D, E had each four blades. In the two screws F, G the pitch was 71¾ inches, and the developed area of the blades was 381 square inches; F had three blades and G six. The arrangement of the blades is shown in the accompanying diagrams, Figs. 1 to 7, looking forwards. The blades are of forged steel, and were pressed by cast-iron molds into their required pitch shape. The bosses are of cast iron, and the blades are keyed on. If the blades, which are inclined slightly aft, are projected on a plane parallel to the axis, their leading and following edges are parallel, as shown in Figs. 8 and 9.


A
B
C
Yacht "Ethel"
a
f
Fig. 1
Fig. 2
Fig. 3
D
Fig. 4
Yacht "Ethel"
Fig. 5
E
Fig. 8
F
Yacht "Ethel"
Fig. 6
G
Fig. 7
Fig. 9
Expanded Blade
p
Fig. 10
q
r
Bradley & Poates, Engr's, N.Y.

The first series A to E, Figs. 1 to 5, consists essentially of two double-bladed propellers, each having its two blades opposite to each other. The stern shafting was made to receive both these propellers at the same time, one being placed immediately in front of the other and touching it (Fig. 8). The forward screw *f* was permanently keyed for the time being on the shaft; while the after screw *a* had seven keyways in the boss, and was tried in seven different positions in reference to the forward screw; five of these positions form the screws A to E, and the two other positions are not considered in this paper.

The second series F and G, Figs 6 and 7, consists of two three-bladed propellers, which were fixed on the shaft in the same manner as those in the first series. It will be noticed that the blades of the second series are narrower or shorter than those of the first; but the thickness at the roots and tips, and also the coefficient of surface friction, are practically the same in both series.

Screw A, Fig. 1, was formed by bringing the leading edge of the after portion *a* into contact with the following edge of the forward portion *f*, thus forming a two-bladed propeller of the ordinary kind; the edges were a good fit, and two correct helicoidal blades were thereby formed. In screw B, Fig. 2, the after portion was placed immediately behind the forward. Screw C, Fig. 3, was formed from screw A, Fig. 1, by turning the after blades forward in the direction of rotation, into the position shown dotted, so that the after blades rotate in advance of the forward; the roots of the leading edges of the forward blades and of the following edges of the after blades are here in the same straight lines, parallel to the axis. In screw D, Fig. 4, the four blades are at right angles to one another; but this is not a propeller of the ordinary kind, because the alternate blades revolve in different planes. Screw E, Fig. 5, was obtained by turning the after portion, backwards in the contrary direction to the rotation, into a position just behind the forward portion, so that the after portion rotates immediately in the wake of the forward portion; this is not unlike an ordinary two-bladed propeller, but having a gap in the middle of the width of each blade. Screw F, Fig. 6,

Table I.

SCREW A, TWO BLADES, PITCH 6¼ INCHES. FIG. 1.

Speed, miles per hour.....	4	5	6	7	7·5	8
Revolutions of screw per minute.....	83.75	109.11	137.91	172.00	190.30	209.88
Slip of screw, per cent....	21.4	24.7	28.4	33.06	35.1	37·3
Indicated Horse Powers..	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
Propulsion.....	1.496	2.591	4.610	8.771	11.733	14.915
Slip of screw.....	0.407	0.849	1.828	4.332	6.346	8.872
Friction of screw blades..	0.069	0.154	0.312	0.605	0 816	1.100
" due to load.....	0.159	0.291	0.547	1.111	1.532	2.018
Net power.....	2.131	3.885	7.297	14.819	20 427	26.905
Friction of engine.....	1.282	1.671	2.113	2.634	2 916	3.215
Total power.....	3.413	5.556	9 410	17.453	23.343	30.120
Mean Pressure per Square Inch.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
Propulsion.....	10.50	13.95	19.64	29.96	36.22	41.75
Slip of screw.....	2.86	4.57	7·79	14.80	19 59	24.84
Friction of screw blades..	0.48	0.83	1.33	2.07	2.52	3 08
" due to load.....	1.11	1.57	2 33	3·79	4·73	5.65
" of engine.....	9.00	9.00	9.00	9.00	9.00	9 00
Total mean pressure.....	23 95	29.92	40.09	59.62	72.06	84.32
Thrust, lb.....	140 2	194.3	288.1	469.9	586.6	699.1
Co-efficient of Performance†.....	85.91	103.07	105.15	90.03	82.79	77.87
Efficiency, per cent.....	70.53	66.69	63.17	59.19	57.44	55.43

* Referred to high-pressure cylinder.

† Coefficient of performance $= \sqrt[3]{\text{displacement in tons}^2} \times (\text{speed in knots})^3 \div$ total I.H.P.

was arranged in a similar manner to screw A, thus forming a three-bladed propeller of the ordinary kind. In screw G, Fig. 7, the six blades are placed at equal angles with one another, and the alternate blades revolve in different planes.

Vessel.—The yacht *Ethel*, on which the experiments were made, is owned by Mr. G. A. Newall, of Bristol, to whom also the author's thanks are due for much valuable assistance and advice. It was built and engined by his firm in 1886. It is 55 feet long and 9 feet beam; the mean draught during the experiments was 3 feet 3 inches, corresponding with a displacement of 18½ tons. There are two saloons on deck, one fore and one aft. The engines are vertical compound surface-condensing of

the ordinary marine pattern. The diameters of the high and low-pressure cylinders are 7 and 14 inches respectively and the stroke is 9 inches; the high-pressure cylinder is placed forward; the piston rods are $1\frac{5}{8}$ inch in diameter. The boiler is placed forward of the engines, and is 5 feet in diameter and 6 feet 6 inches in length; the working pressure is 100 pounds per square inch. Before commencing the experiments, the engines, though in good condition, were thoroughly overhauled. The indicator pipes were fitted carefully, and with as few bends as possible. The fire-grate area of the boiler was enlarged, as were also the areas of the uptakes and funnel, in order to enable the boiler to maintain a constant pressure of steam at all speeds.

Table II.

SCREW B, FOUR BLADES, PITCH 64¼ INCHES. FIG. 2.

Speed, miles per hour	4	5	6	7	7·5	8
Revolutions of screw per minute	83.75	109.11	136.98	172.10	192.33	211.99
Slip of screw, per cent	21.4	24.7	27.9	33.07	35.8	37.9
Indicated Horse Powers	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
Propulsion	1.496	2.591	4.613	8.775	11.731	14.923
Slip of screw	0.407	0.849	1.785	4.335	6.541	9.107
Friction of screw blades	0.069	0.154	0.307	0 605	0.846	1.134
" due to load	0.159	0.291	0.543	1.112	1.550	2.040
Net power	2.131	3.885	7.248	14.827	20.668	27.204
Friction of engine	1.282	1.671	2.098	2.636	2.946	3.247
Total power	3.413	5.556	9.346	17.463	23.614	30.451
Mean Pressures per Square Inch.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
Propulsion	10.50	13.95	19.79	29.97	35.84	41.36
Slip of screw	2.86	4.57	7.65	14.79	19.98	25.24
Friction of screw blades	0.48	0.83	1.32	2.07	2.58	3.14
" due to load	1.11	1.57	2.33	3.79	4.74	5.66
" of engine	9.00	9.00	9.00	9.00	9.00	9.00
Total mean pressure	23.95	29.92	40.09	59.62	72.14	84.40
Thrust, lb	140.2	194.3	288.3	469.9	586.5	699.5
Co-efficient of performance †	85.91	103.07	105.87	89.98	81.85	77.03
Efficiency, per cent	70.53	66.69	63.64	59.18	56.75	54.86

* Referred to high-pressure cylinder.

† Coefficient of performance $= \sqrt[3]{\text{displacement in tons}^2} \times (\text{speed in knots})^3 \div \text{total I.H.P.}$

Table III.

SCREW C, FOUR BLADES, PITCH 6¼ INCHES. FIG. 3.

Speed, miles per hour.....	4	5	6	7	7.5	8
Revolutions of screw per minute	83.75	109.11	136.98	167.76	186.20	203.00
Slip of screw, per cent...	21.4	24.7	27.9	31.3	33.7	35.2
Indicated Horse Powers...	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
Propulsion	1.496	2.591	4.613	8.779	11.728	14.911
Slip of screw.....	0.407	0.849	1.785	4.000	5.961	8.099
Friction of screw blades..	0.069	0.154	0.307	0.562	0.768	0.995
" due to load.....	0.159	0.291	0.543	1.081	1.496	1.946
Net power.....	2.131	3.885	7.248	14.422	19.953	25.951
Friction of engine.....	1.282	1.671	2.098	2.570	2.852	3.109
Total power.....	3.413	5.556	9.346	16.992	22.805	29.060
Mean Pressures per Square Inch.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
Propulsion.....	10.50	13.95	19 79	30.75	37.01	43.16
Slip of screw.....	2.86	4.57	7.65	14.01	18.81	23 44
Friction of screw blades..	0.48	0.83	1.32	1.96	2.42	2.88
" due to load.....	1.11	1.57	2.33	3.79	4.72	5.63
" of engine	9.00	9.00	9.00	9.00	9.00	9.00
Total mean pressure.....	23.95	29.92	40.09	59.51	71.96	84.11
Thrust, lb.....	140.2	194.3	288.3	470.3	586.4	698.9
Co-efficient of performance†.....	85.91	103.07	105 87	92.47	84.94	80.71
Efficiency, per cent.....	70.53	66.69	63.64	60 87	58.77	57.45

* Referred to high-pressure cylinder.

† Coefficient of performance $= \sqrt[3]{\text{displacement in tons}^2} \times (\text{speed in knots})^3 \div$ total I.H.P.

Mode of Experiment.—The trials were made on the Avon, on the regatta course at Saltford, about midway between Bath and Bristol. This was an exceptionally suitable place, as there is a perfectly straight run of nearly a mile, the sides of the course are parallel, and the width and depth of the water ample, and it is also well shaded. The width is about 100 feet, the depth about 14 feet, and the sectional area of the river about 1,200 square feet, while the greatest immersed section of the yacht is 20.4 square feet for the displacement of 18½ tons. During the time the runs were made the course was entirely free from traffic; the water was smooth, and the speed and depth of the current varied but little. The experiments were made on a base line of 3,400

feet, or 0.64 mile. Two upright poles were erected at each end of the base line, one twenty yards behind the other in a line at right angles to the stream. Each screw was tried at six successive speeds, namely, 4, 5, 6, 7, 7½ and 8 miles per hour, as nearly as possible. For each speed the yacht was run over the course six times, three in each direction, and the mean was taken. The supply of steam was regulated by the throttle valve, which was locked for each experiment, the reversing gear being fully open. Two indicators were employed, one on each cylinder, and each was connected with both top and bottom. Three sets of diagrams were taken during each run, or eighteen for each experiment, making altogether 72 single diagrams. The same set of springs, excepting a broken one, were employed throughout the experiments. The counter was placed in a convenient position on deck, and motion was given to it by a rod from the engine crosshead; it could be stopped or started without in-

Table IV.

SCREWS D AND E, FOUR BLADES, PITCH 6¼ INCHES. FIGS. 4 AND 5, AND 19 TO 23.

Screw	D	D	D	D	D	D	E
Speed, miles per hour	4	5	6	7	7.5	8	8
Revolutions of screw per minute	83.75	109.11	136.98	169.30	188.31	208.00	218.00
Slip of screw, per cent	21.4	24.7	27.9	31.9	34.4	36.7	39.6
Indicated Horse Powers.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
Propulsion	1.496	2.591	4.613	8.776	11.736	14.902	14 897
Slip of screw	0.407	0.849	1.785	4.110	6.154	8.639	9.770
Friction of screw blades	0.069	0.154	0.307	0.577	0.795	1.070	1.233
" due to load	0.159	0.291	0.543	1.091	1.515	1.995	2.100
Net power	2.131	3.885	7.248	14.554	20.200	26.606	28.000
Friction of engine	1.282	1.671	2.098	2.593	2.884	3.186	3.338
Total power	3.413	5.556	9.346	17.147	23.084	29.792	31.338
Mean Pressures per Square Inch.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
Propulsion	10.50	13.95	19.79	30.46	36.62	42.10	40.16
Slip of screw	2.86	4.57	7.65	14.27	19.20	24.40	26.33
Friction of screw blades	0.48	0.83	1.32	2 00	2.48	3.02	3.32
" due to load	1.11	1.57	2.33	3.79	4.72	5.64	5.66
" of engine	9.00	9.00	9.00	9.00	9.00	9.00	9.00
Total mean pressure	23.95	29.92	40 09	59.52	72.02	84.16	84.47
Thrust, lb	140.2	194.3	288.3	470.1	586.8	698.5	698.3
Co-efficient of performance†	85.91	103.07	105.87	91.63	83 72	78.73	74.80
Efficiency, per cent	70.53	66.69	63.64	60.33	58.09	56 00	53.21

* Referred to high-pressure cylinder.

† Coefficient of performance $= \frac{\sqrt[3]{\text{displacement in tons}^2} \times (\text{speed in knots})^3}{\text{total I.H.P.}}$

Table V.

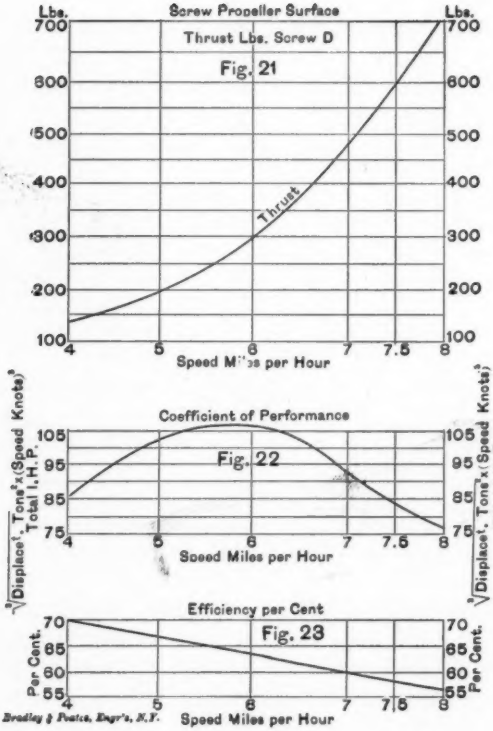
SCREW F, THREE BLADES, PITCH 71¾ INCHES. FIG. 6.

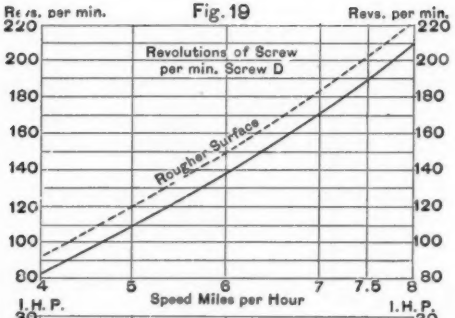
Speed, miles per hour	4	5	6	7	7·5	8
Revolutions of screw per minute	76.4	100 45	127.31	154.90	172.48	192.21
Slip of screw, per cent	23.0	26.7	30.6	33·5	36.0	38.7
Indicated Horse Powers	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
Propulsion	1.500	2.594	4.616	8.771	11.739	14.918
Slip of screw	0.448	0.945	2.036	4 418	6.602	9.457
Friction of screw blades	0.056	0.127	0 259	0.469	0.646	0 894
" due to load	0.163	0.297	0.560	1.107	1.539	2.049
Net power	2.167	3.963	7.471	14.765	20.526	27.318
Friction of engine	1.170	1.538	1.950	2.372	2.642	2.945
Total power	3.337	5.501	9 421	17.137	23.168	30.263
Mean Pressures per Square Inch.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
Propulsion	11.54	15.17	21.30	33.27	39.99	45.61
Slip of screw	3.45	5.52	9.40	16.76	22 49	28.91
Friction of screw blades	0.43	0.75	1.20	1.78	2.20	2.73
" due to load	1.25	1.74	2.58	4.19	5.24	6.26
" of engine	9.00	9.00	9.00	9.00	9.00	9.00
Total mean pressure	25.67	32.18	43.48	65.00	78.92	92.51
Thrust, lb	140.6	194.5	288.5	469.9	586.9	699.3
Co-efficient of performance†	87.86	104.10	105.06	91.69	83.42	77.52
Efficiency, per cent	69.22	65 45	61.78	59.40	57.19	54.61

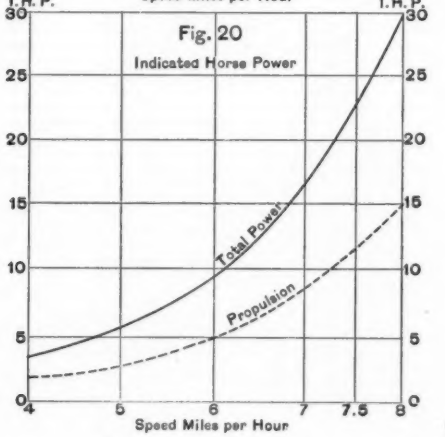
* Referred to high-pressure cylinder.

† Coefficient of performance $= \sqrt[3]{\text{displacement in tons}^2} \times (\text{speed in knots})^3 \div \text{total I.H.P.}$

terfering with the engines. The working pressure in the boiler was constant at 100 pounds per square inch throughout the experiments. It was found at first that in the single instance of full speed the boiler pressure was inclined to vary somewhat. This objection was overcome by the aid of forced draft, derived from a small fan, which was driven off the screw shafting. The condenser vacuum was constant at 27 inches of mercury. In making the runs, while the vessel was passing the two ranging poles at either extremity of the base line, a chronometer and the counter were started or stopped, as the case might be, at the moment when the two poles were in a straight line with a point


Lbs.
Screw Propeller Surface
Lbs.
700
Thrust Lbs. Screw D
Fig. 21
600
500
400
Thrust
300
200
100
4
5
6
7
7.5
8
Speed Miles per Hour
Coefficient of Performance
105
95
85
75
Fig. 22
Speed Miles per Hour
$\sqrt[3]{\text{Displace}^t.\ \text{Tons}^2 \times (\text{Speed Knots})^3}$ / Total I.H.P.
Efficiency per Cent
70
65
60
55
Per Cent.
Fig. 23
Speed Miles per Hour
Bradley & Peaco, Engr's, N.Y.


Revs. per min.
Fig. 19
Revs. per min.
220
200
180
160
140
120
100
80
Revolutions of Screw per min. Screw D
Rougher Surface
4
5
6
7
7.5
8
Speed Miles per Hour


I.H.P.
Fig. 20
Indicated Horse Power
I.H.P.
30
25
20
15
10
5
0
Total Power
Propulsion
4
5
6
7
7.5
8
Speed Miles per Hour

on deck. The time was taken to a third of a second. In order to insure having the same weight on board during the runs, the coal was weighed as it was used, and its weight was replaced at the end of each experiment. The same steersman and engineer were employed throughout the experiments, and the readings were taken by a staff of four observers.

The total indicated horse power and the revolutions per minute were obtained exactly for the speeds of 4, 5, 6, 7, 7½ and 8 miles per hour, by plotting the experimental results and drawing fair curves through them, as shown in Figs. 19 to 23 for screw D, and then scaling off the quantities from the curves corresponding with these speeds. The equivalents of these speeds in knots

Table VI.

SCREW G, SIX BLADES, PITCH 71¾ INCHES. FIG. 7.

Speed, miles per hour	4	5	6	7	7.5	8
Revolutions of screw per minute	76.4	100 00	124.90	151.00	168.08	187.34
Slip of screw, per cent	23.0	26.4	29.3	31.8	34.3	37.2
Indicated Horse Powers	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.	I.H.P.
Propulsion	1.500	2 594	4 616	8.771	11.740	14.922
Slip of screw	0.448	0.930	1.909	4.089	6.133	8.838
Friction of screw blades	0.056	0.126	0.231	0.434	0.597	0.828
" due to load	0.163	0.295	0.547	1.077	1.497	1.993
Net power	2 167	3.945	7.303	14.371	19.967	26.581
Friction of engine	1.170	1.532	1.913	2.313	2.574	2.869
Total power	3.337	5.477	9.216	16.684	22.541	29.450
Mean Pressures per Square Inch.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
Propulsion	11.54	15.25	21.71	34.12	41.04	46 84
Slip of screw	3.45	5.46	9.00	15.91	21.44	27.72
Friction of screw blades	0.43	0.74	1.08	1.70	2.07	2.60
" due to load	1 25	1.73	2.57	4.19	5.26	6.25
" of engine	9.00	9.00	9.00	9.00	9.00	9.00
Total mean pressure	25.67	32.18	43.36	64.92	78.81	92.37
Thrust, lb	140.6	194.5	288.4	470.0	586.9	699.4
Co-efficient of performance†	87.86	104.55	107.37	94.18	85.74	79.51
Efficiency, per cent	69.22	65 75	63.20	61 03	58.80	56.14

* Referred to high-pressure cylinder.

† Coefficient of performance $= \sqrt[3]{\text{displacement in tons}^2} \times (\text{speed in knots})^3 \div \text{total I.H.P.}$

are 3.47, 4.34, 5.20, 6.07, 651 and 6.94 knots. The results of the seven screws, A to G, together with the analysis, are given in Tables 1 to 6.

The efficiency of the screws has been examined by the method employed by Mr. Isherwood, the late Engineer in Chief of the Navy Department of the United States. It is not unlike the method proposed by the late Dr. Froude (Inst. Naval Architects, 1876, page 167), whereby he calculated the indicated thrust from the piston pressures, and gave a number of elements which must be subtracted from the indicated thrust in order to ascertain the ship's true resistance. As the screws in each series of the present experiments have exactly the same pitch, diameter and physical surface, they have therefore the same coefficient of friction; and although some of the quantities taken in the analysis are perhaps only approximately correct, such as the coefficient of friction, the experiments are nevertheless thoroughly suitable for comparison with one another, and free from the objections which generally apply to propellers having slightly different dimensions, although meant to be the same.

For convenience the mean pressures of the compound engine have been referred to the high-pressure cylinder. The effective mean pressure is therefore the actual mean pressure P in the high-pressure cylinder, together with the mean pressure p in the low pressure multiplied by the ratio 4.083 of their capacities; that is, the effective mean pressure is $P + p \times 4.083$.

The indicated horse power developed by the engine has been resolved into the following five constituent parts:

1. The power to overcome the friction of the unloaded engine, which includes the duty of the air pump and the feed pump.
2. The power to overcome the friction due to the working load.
3. The power to overcome the friction of the screw blades.
4. The power expended in the slip of the screw.
5. The power necessary for the propulsion of the vessel.

The net horse power applied to the shaft is obtained by deducting from the total indicated horse power the power to overcome the friction of the engine. The steam pressure necessary

to overcome the friction of the engine was equal to 9 pounds per square inch, and was constant at all speeds. Following Mr. Isherwood (Inst. Civil Engineers, Vol. CII, page 163), the friction due to the working load was taken at $7\frac{1}{2}$ per cent. of the net power; and the blade friction at 0.45 pound per square foot of blade surface when moving at a velocity of 10 feet per second, and for other speeds in the ratio of the squares of those speeds to the square 10. If the first three of the above quantities, Nos. 1 to 3, are subtracted from the total indicated horse power, there remains a quantity, the sum of Nos. 4 and 5, absorbed in the action and reaction of the screw, which is the actual power applied to the screw less the surface resistance of the blades. From this remainder was subtracted the slip No. 4, and the final remainder was then the power required to propel the vessel, or the useful work (No. 5). This final power divided by the net power is a measure of the efficiency of the screw. The respective steam pressures may be calculated from the powers.

The thrust was obtained from the following expression: thrust in pounds $\times$ speed of vessel in feet per minute $=$ 33,000 $\times$ horse power expended in propulsion of vessel.

Examination of Results.—It will be seen that screws C and G were the most efficient in the first and second series respectively. Screws B, C and D were exactly equal for all speeds up to six miles per hour. Screw A was also equal to them for speeds up to five miles per hour. Screw C, the most efficient, ranged from equality at these speeds up to 2.02, 2.59, 1.45 and 4.24 per cent. greater efficiency at the maximum speed than the screws A, B, D and E respectively. Screw G was more efficient than screw F at all speeds except the minimum, when they were equal; the difference in efficiency was 1.53 per cent. at the maximum speed of eight miles per hour. It is not unreasonable to suppose that, if a position of blades had been tried in the second series similar to that of screw C in the first, a greater efficiency would have been obtained. It will be seen that at the lower speeds the results of the screws in each series were not affected either by the number or by the position of the blades. The blades of screws

B and E evidently affected the water injuriously for each other at the higher speeds.

The screws attained an efficiency of about 70 per cent. at the minimum speed, with a slip of about 22 per cent., and afterwards fell off with further increase of slip. The screws are in themselves bad, inasmuch as their maximum efficiency should have been at the working speed of about $7\frac{1}{2}$ miles per hour, but the screw space was insufficient to allow for more blade area. This, however, will not in the slightest affect the object of the experiments.

In order to try the effect of friction of the blades, screw D was tested with a much rougher surface, presenting if anything greater roughness than that of very coarse sand. The revolutions are plotted in Fig. 19. The loss of power due to the increased surface resistance was twofold; for, in addition to the extra pressure necessary to rotate the screw at the same number of revolutions, the slip was greater, showing that the transverse component or rotary motion of the water was increased, thereby reducing the sternward or thrust component.

From figures in the following table it is seen that the sum of the mean pressures employed in the propulsion and slip of the screws is practically equal for the screws in each series for the same speeds; in other words, the turning moments necessary to maintain the rotation of the screws, when going at the same speed, are equal; therefore it may be clearly laid down that with screws of equal pitch and equal area of blades, when propelling a vessel at a certain speed, the pressure on the pistons will be equal, whatever may be the state of the water in which the screws work, and the number of revolutions and therefore the slip; for the revolutions will regulate themselves to the existing circumstances of the liquid fulcrum against which the blades act. That such is the case is seen from the fact that, while the pressures have remained constant, the revolutions have varied to a considerable extent.

SUM OF THE PRESSURES OF SLIP AND PROPULSION IN POUNDS PER SQUARE INCH REFERRED TO HIGH-PRESSURE CYLINDER.

Miles per hour.	4	5	6	7	7½	8
Screw.	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*	*lb.*
A	13.36	18.52	27.43	44.76	55 81	66.59
B	13.36	18.52	27.44	44 76	55.82	66.60
C	13.36	18.52	27.44	44 76	55.82	66.60
D	13.36	18.52	27.44	44.73	55 82	66.50
E	.....	.....	.....	.....	.....	66.49
F	14.99	20.69	30.70	50 03	62 48	74.52
G	14.49	20.71	30.71	50.03	62.48	74.52

On comparing the above pressures in the first and second series, they are found to vary in the direct ratio of the pitch, namely, as 1.00 to 1.12, thus showing that for equal speed the turning moment varies in the direct proportion of the pitch. This law is fully demonstrated in Mr. Blechynden's paper on the reaction and efficiency of the screw propeller (North-East Coast Institution of Engineers and Shipbuilders, 1887, vol. iii, page 179); in which he also says the effect of the surface is the same, irrespective of the number of blades into which it is divided. On this point he does not appear to have made any experiments, but accepts the results of Mr. Isherwood, which, however were not sufficiently extended to warrant the assumption.

Discussion of Results.—The results show that, the kind and aggregate extent of surface remaining the same, it is advantageous to increase the number of blades. Hence the blades will become proportionately narrower, and this is undoubtedly the cause of increased efficiency; that is, narrower blades are proportionately more effective than wider ones. For when the two-bladed and three-bladed propellers were divided respectively into four blades at right angles to one another (screw D) and into six blades (screw G), the slip was reduced for the same speed with exactly the same steam pressure; consequently the efficiency is greater. This conclusion appears to be somewhat contrary to prevailing opinions, but the author is not acquainted with any trials of propellers having six or more blades. The

above remark refers only to smooth water; in a rough sea the advantage of many blades appears obvious. In his "Experiments in Aerodynamics," published by the Smithsonian Institution, Washington, 1891, Mr. S. P. Langley has shown (page 106) that with horizontal planes moving horizontally in air the time in falling is greater for those planes whose extension from front to back is small compared with their length measured perpendicularly to the line of advance; in other words, narrower planes have less slip than wider ones.

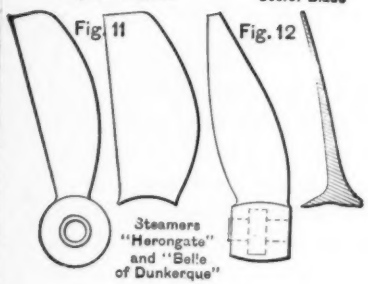
It does not seem difficult to find an explanation for the foregoing result. The normal pressure on a blade moving obliquely through water does not occur at the center of the width, but at a position about one-fourth or one-third of the width from the leading edge. If a sheet of paper or metal is moved through air or water with its surface inclined to the direction of motion, the leading edge will suffer considerable deflection compared with the following, and will tend to assume a position at right angles to the line of motion. The leading face of a propeller blade is thus more effective than the following. In a paper read before the Institution of Naval Architects in 1890, Mr. Howden has noticed this important fact, for he says (page 237), "it is evident that the efficiency of the blade decreases towards the after edge. The first 6 inches on the entering side are more effective than the last 6 inches on the after side." It is thus chiefly by the leading side of the blade that the inertia of the water is overcome and its acceleration produced, probably in the form of an impact, thereby giving the water a greater velocity than that of the screw itself. This would be a cause for apparent negative slip, the water flowing through the propeller faster than is accounted for by the product of the pitch and revolution. There must then be a region of reduced pressure on the working face of the blade. That there is a vacuum in the wake of a blade, caused by the slip and by the bottom of the blade displacing the water, is proved by the performance of screw E; for with exactly the same steam pressure the revolutions increased to a marked extent, showing that the after section was exerting little or no thrust.

Expanded Blade

Fig. 11

Sec. of Blade

Fig. 12



Steamers "Herongate" and "Belle of Dunkerque"

Tug "Frank Stanley"

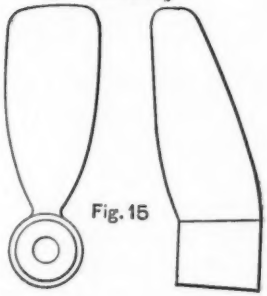


Fig. 15

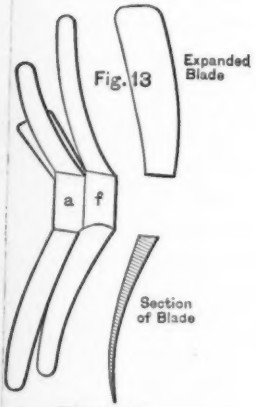


Fig. 13

Expanded Blade

Section of Blade

Steamers "Herongate" and Belle of Dunkerque"

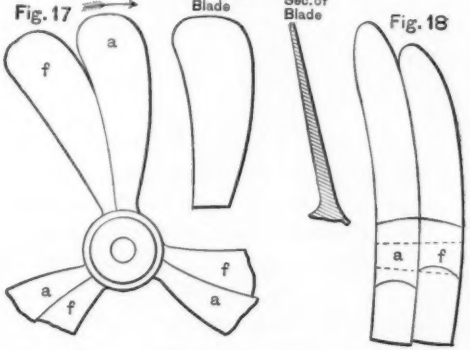
Tug "Frank Stanley"

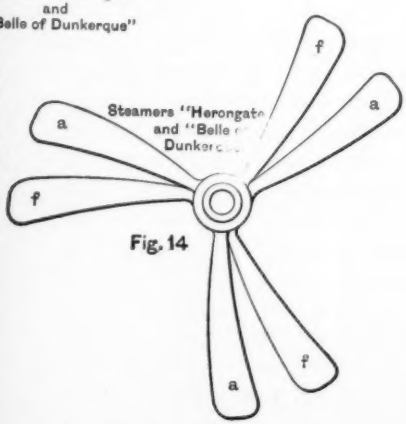
Fig. 17

Expanded Blade

Sec. of Blade

Fig. 18





Steamers "Herongate" and "Belle of Dunkerque"

Fig. 14

Tug "Frank Stanley"

Expanded Blade

Sec. of Blade

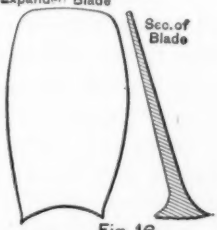


Fig. 16

Bradley & Bates, Engr's, N.Y.

It may be well to mention the case of a propeller 10 feet in diameter, which was brought under the author's notice after some five or six months' continued voyaging to the Mediterranean. In Fig. 10 is shown an expanded view of the working face, of which the portion *p* on the leading edge was clean scoured; the part *r* was covered with marine growth some inches long, while the part *q* on the after edge had a short growth; the greater part of the back of the blade was covered with grass. This seems to show that only a small part of the area of the blade was effective, and affords an argument in favor of narrow blades.

In accounting for the superiority of screw C, it will be noticed that the blades of each portion, when the vessel is in motion, necessarily cut into water undisturbed with respect to the screw, which was probably not the case with the other screws, whose blades injuriously disturbed the water for one another. Moreover, the spent water from the forward blades of screw C, being discharged with a velocity at the back of the after blades, destroys the partial vacuum behind the after blades, thus augmenting the thrust by preventing the water from tending to run round to the back of the after blades.

Practical Trials.—Previous to carrying out the above experiments some trials were made with propellers on the principle of screw C, which is the form invented by Mr. Dickinson, by whose permission the results are here given, being of interest for comparison with those subsequently obtained from the experiments. After several successful trials on steam launches, two vessels belonging to Messrs. Weatherley, Mead & Hussey, of London, the *Herongate* and the *Belle of Dunkerque*, were kindly offered by the owners for further trial of the screw. These vessels are sister ships in every respect, built in 1887 by Messrs. Short Brothers, and engined on the triple-expansion plan by Mr. John Dickinson, of Sunderland, each of 525 tons register. The ordinary propeller, shown in Figs. 11 and 12, was of cast iron, 10 feet 9 inches in diameter, 15 feet pitch, with four blades and 32 square feet area; the Dickinson screw, shown in Figs. 13 and 14, was

of cast steel, 10 feet 6 inches in diameter and 15 feet pitch, made as a double three-bladed screw with six blades and 30 square feet area.

Table VII.

VOYAGES OF TWO STEAMERS TO SPANISH PORTS IN THE MEDITERRANEAN WITH ORDINARY FOUR-BLADED PROPELLER (FIGS. 11 AND 12) AND DICKINSON'S SIX-BLADED PROPELLER (FIGS. 13 AND 14).

Steamer	*Belle of Dunkerque.*		*Herongate.*			
Propeller	Ordinary propeller.		Ordinary propeller.		Dickinson screw.	
Voyage	Cardiff to Alicante.	Cardiff to Almeria.	London to Denia.	London to Valencia.	London to Cullera.	Cardiff to Tarragona.
Distance run, knots	1,458	1,288	1,611	1,478	1,632	1,660
Time, hours	165	149	184	184	181	196
Speed, knots	8.836	8.640	8.755	8.032	9.016	8.470
Revolutions of screw per minute	83	84	85	79	74	73
Boiler pressure, lbs. per sq. in.	152	153	155	143	150	155
Vacuum, in. of merc'y.	25	25	24	24	25	25
Draught of steamer	12′ 7″	12′ 10″	12′ 8½″	12′ 9″	12′ 8½″	12′ 11½″
Coal burnt in twenty-four hours, tons	7½	7½	7	6½	7	6
Knots run per ton of coal	28.27	27.76	30.02	29.66	30.91	33.88
Slip of screw, per ct.	28.0	30.5	30.4	31 3	17.7	21 6

In Table VII (above) is given a record of the first two deeply-laden voyages of these two vessels to Spanish ports in the Mediterranean, the ordinary propellers being exactly the same on both vessels. The following are the mean results:

COMPARISON OF ORDINARY FOUR-BLADED AND DICKINSON SIX-BLADED PROPELLER, AVERAGE RESULTS OF TABLE VII.

Steamer	*Belle of Dunkerque.*	*Herongate.*	
Propeller	Ordinary.	Ordinary.	Dickinson.
Speed, knots	8.738	8.393	8.743
Revolutions of screw per minute	83.5	82.0	73.5
Boiler pressure, pounds per square inch	152.5	149.0	152.5
Vacuum, inches of mercury	25 0	24.0	25.0
Draught of steamer, feet and inches	12 8½	12 8¾	12 10
Coal burnt in twenty-four hours, tons	7½	6¾	6½
Knots run per ton of coal	28.0	29.8	32.4
Slip of screw, per cent.	29 2	30 8	19.6
Allowance for foul bottom, knots per hour	0.33	.....	0.50

When running with the Dickinson propeller the condition of the *Herongate* was very foul. On the first voyage, from London to Cullera, she started with a three months' growth on a coating of tar. On her return she made a voyage to the Baltic, after which she started on her voyage from Cardiff to Tarragona with a growth of five months and a half. Returning thence, she was found to be extremely foul, being covered from keel to water line with grass three inches to four inches long. She was therefore cleaned and painted, and thus commenced with a clean bottom her voyage from London to Denia. On her return, having been aground, she was again docked and painted with anti-fouling composition before starting on her next voyage from London to Valencia. The *Belle of Dunkerque* had a growth of one month and two months and a half on commencing her first and second voyage respectively.

In the navigation of the *Herongate* it was found that the Dickinson propeller afforded greater command of the vessel, through improved steering and greater rapidity in evolution, and power of stopping and starting more promptly, and there was a reduction in vibration. It was also reported that, when running high on a ballast trim, with the propeller not thoroughly submerged, the efficiency was even more pronounced, inasmuch as the screw possessed a firmer grip of the water, and held the vessel steady to the wind.

The *Herongate* was also run from Upper Blyth buoy in Sea Reach (the second reach below Gravesend) to the North Sand Head lightship (the North Goodwin lightship), first with the ordinary and afterwards with the six-bladed propeller. These trials were conducted by Mr. George R. Dunell, and the following results were obtained:

COMPARATIVE TRIALS OF THE *HERONGATE* WITH ORDINARY FOUR-BLADED AND DICKINSON SIX-BLADED PROPELLER.

Propeller	Ordinary	Dickinson
Duration of trial, hours and minutes	5 40	4 27
Distance by log, knots	50	42
Speed, knots	8.82	9.43
Revolutions of screw, per minute	75·3	73·3
Boiler pressure, pounds per square inch	129.2	150.6
Vacuum, inches of mercury	25.6	25.6
Mean total indicated horse power	328.6	389.3
Displacement, tons	1107	1112
Slip of screw, per cent	20.8	13.0
Coefficient of performance	223	232

*Coefficient of performance $=$

$$\frac{\sqrt[3]{\text{Displacement in tons}^2} \times (\text{speed in knots})^3}{\text{Mean total I.H.P.}}$$

It must be remembered in comparing these two performances that the vessel had a foul bottom when tried with the Dickinson propeller, whilst when running with the ordinary propeller she was just out of dock after cleaning.

Trial on Tugboat.—On the completion of the experiments at Saltford, the author carried out a trial on the tugboat *Frank Stanley*, which was kindly placed at his disposal by Captain Roberts, of Bristol. The object of this trial was more especially to ascertain how far a screw with narrow blades was suitable for general towing. The ordinary four-bladed propeller of the tug is shown in Figs. 15 and 16; the Dickinson screw employed is shown in Figs. 17 and 18, having six blades in two portions. The tug is 70 feet in length, and 14 feet beam, with a draught of 5 feet forward and 7½ feet aft. Both screws were of cast iron and 6 feet in diameter; the ordinary four-bladed had a pitch of 9 feet and an expanded blade area of 13.58 square feet; the pitch of the Dickinson screw was 9 feet 6 inches, and its blade area 12.12 square feet. The following were the results of the trial:

COMPARATIVE TRIALS OF THE TUG *FRANK STANLEY* WITH ORDINARY FOUR-BLADED AND DICKINSON SIX-BLADED PROPELLER.

Propeller	Ordinary.	Dickinson.
Speed, knots	10.8	11.1
Revolutions of screw per minute	136.7	128.0
Boiler pressure, pounds per square inch	58.5	57 9
Vacuum, inches of mercury	15.0	15 0
Slip of screw, per cent	11.1	7·5

After three months' continuous work in all seas and weathers, Captain Roberts reported as follows respecting the towing qualities of the six-bladed propellers: "While the towing power of the tug is greater, by an amount, I should think, approaching six or seven horse power, the action of the propeller on the water is also smoother, and it appears to have a firmer grip of the water, which is especially noticeable when getting under weigh, the required speed being obtained almost immediately on starting the engine."

These practical trials on the *Herongate* and the *Frank Stanley* form a strong corroboration of the somewhat more scientific trials at Saltford, demonstrating the superiority of narrow blades, which is obtained by increasing their number and arranging them on the principle of the screw C, Fig. 3. The slip is thereby reduced, with even less blade area, as is seen from an examination of the results. The practical objections are overcome by making the propeller in two portions, which are more easily manufactured and handled, and can be fitted on without the often tedious necessity of withdrawing the stern shafting. All that is required in case of accident is to replace the injured portion. This principle applies advantageously to vessels of light draught; for by increasing the number of portions the diameter of the screw may be proportionately diminished and the required blade area obtained with a uniform pitch. That the steering is much improved is obvious, in consequence of the greater distribution of water discharged against the rudder, whereby also the vibration is reduced to a considerable extent.

In conclusion, it will be understood that the author does not wish to advocate a particular kind of screw propeller, or even to assert that any one form is the best, as there are so many considerations affecting the question; but he desires rather to draw attention to several important points bearing upon the subject.

Discussion.

(As Reported by "Engineering.")

Mr. R. E. Froude, the Chief of the Admiralty experimental works at Haslar, was the first speaker. He bore testimony to

the value of the paper, and the scientific accuracy with which it had been prepared. One great advantage the author possessed was the excellent course on which he had run the trials, where the freedom from current eliminated a very fruitful source of errors. In experiments of this nature the measurement of the speed of vessel was quite the most important thing, and tidal influences made it often difficult to determine this with absolute certainty. But although Mr. Froude allowed the accuracy of the observations he was prepared to criticise some of the author's conclusions, which he thought were often too precise, considering the foundations upon which they were based. He did not question that analyses were based on sound principles, but he thought they were drawn to too fine a point. For example, in the elimination of the blade friction no one knew what the influence of that factor might be; an approximation only could be reached. This was well enough for some purposes, but at best it was a crude approximation. The author had stated that the screws tried had the same coefficient of friction and admitted that some of the quantities taken in the analysis were perhaps only approximately correct, including the coefficient of friction. In Mr. Froude's opinion the allowance in this respect was too low. In a good screw, the speaker stated, with a slip ratio giving maximum efficiency, the power consumed in edgewise resistance should be equal to the slip. That had been proved by the late Dr. Froude, but in the cases cited the edgewise resistance varied from one-sixth to one-eighth of the slip only.

Also in estimating the power to be apportioned to the influence of slip, not the apparent slip but the true slip should be taken. It should be remembered that in estimating these matters minute differences were of importance in calculating the element of blade friction.

The next element was engine friction. That he understood the author had ascertained by experiment, but the friction of the load was always more or less in the nature of a guess, yet that was one of the factors used in arriving at the author's conclusions. As a whole it might be constant, but it fluctuated considerably; quite enough to influence the results, although in the

paper it was given as constant. When he looked at the conclusions given in the tables at the end of the paper and found them worked out to two places of decimals, he felt afraid they were rather too good to be true. The figures in the tables as to speed were doubtless unusually accurate, and the steam pressures were likewise accurate within the limits of the indicator diagrams, but between those lines were other lines which were not quite so certain. The observed results were such as he would expect within the limits of error, and the figures were to be accepted, omitting the element of edgewise resistance.

Mr. Froude referred also to the resistence varying in longer and shorter surfaces. This was a thing which he did not remember the time when he did not know; but the tendency of water to "short-circuit" over the end of the blade, a fact which Mr. Thornycroft had pointed out, should be taken into consideration. There was a layer of water moving with the blade thicker or thinner according to the width of the blade, but at the end the water escapes over the edge and follows the course of least resistance. If the designer of the propeller were not limited by the question of structural strength there would be hardly a limit to which the narrowing of the blade might not be carried, but it had to be made thick enough to bear up against the thrust, and hence there was a point at which bluntness would become excessive in order to meet structural necessities. Mr. Froude considered that the author's experiments exaggerated the advantages of multiplying the blades, as they did not allow the importance due to edgewise resistance. The experiments made by the author were very accurately carried out, and, in Mr. Froude's opinion, went to show that it was not advantageous to multiply the number of blades beyond that which was usual.

Mr. Thornycroft said that it had been suggested that the experiments of the author should be carried out on a larger scale. This Mr. Thornycroft did not consider advisable, for—putting other reasons on one side—life was not long enough. The late Mr. Froude had come to that conclusion with regard to ships, and he had therefore contented himself with experimenting on

models. He and Mr. Barnaby had spent some time in testing model screws. They had had a steam launch especially fitted for the purpose, and had given two years to an endeavor to ascertain one point in connection with screw design; and yet they had used little models that could be slipped on and off during the course of the experiment in a minute or two. If the trials had been on the larger scale suggested, an enormous time would have been taken up in docking the ship to change the propeller, and the experiments could not have been carried out within the compass of one man's experience. With regard to the Dickinson propeller, it was always difficult to find out whether one screw was better than another, but his own opinion was that the now ordinary form was the most advantageous. In the matter of efficiencies it might be taken fundamentally that the fewer the blades the better, but there were other limiting conditions. For his own part, he would make a propeller with one blade if he could, but three were often used with advantage, for reasons that were manifest.

Mr. Sydney Barnaby considered that the experiments conducted by Mr. Walker were well worthy of attention as they had been very carefully conducted, but he did not attach importance to the reports of captains of vessels at the end of the paper. He thought it was not difficult to account for the slight superiorty of the Dickinson screw at the higher speeds of the launch. At these speeds especially the propellers were all much too small, and when the diameter was restricted it was always advisable to give four blades in order that the surface required might be well distributed. This could not be efficiently done with two wide blades as in screw A, especially if the pitch was uniform as was the case with all those tried on the launch. Although uniform pitch was good enough in very narrow blades, wide ones required to be curved, that is required an increasing pitch from forward to aft in order that they should fit the contracting column of water passing through them, and he therefore thought that the comparison between screws A and C was not a fair one. With regard to Mr. Walker's remarks upon apparent negative slip he, Mr.

Barnaby, had stated in his paper read before the Institution of Civil Engineers in 1890, that he attributed it to the fact that the race receded from the screw with a greater velocity than was accounted for by the product of pitch and revolution, but he did not agree that the velocity was produced by the "impact" of the blade against the water as suggested by the author. Impact could not be got in the case of a submerged propeller.

Mr. Dunell was the next speaker. He said that the experiments of which particulars are given by the author were but a part of a series the speaker had made for Mr. Dickinson. There is no doubt that the *Herongate* did somewhat better with the Dickinson propeller than with her original propeller; but that simply proved that for an individual ship one propeller was better than another, and it was altogether insufficient ground upon which to found a principle. The particulars of the *Herongate's* trials had no value, the speaker said, from a scientific point of view. They were only a comparative test of one propeller against another, and they might be set against another set of trials the speaker had made for Mr. Dickinson later on. These were with a second-class torpedo boat which Mr. Yarrow very kindly lent to Mr. Dickinson for the purpose; and not only that, but gave the services of a crew, stores and coal. A special Dickinson propeller was made, having blade area, diameter and pitch to correspond with the design adopted by Mr. Yarrow in his ordinary torpedo boat practice. The result was that the Yarrow propeller gave a speed of 17.112 knots, and the Dickinson propeller 15 knots. A word should be said in explanation of the figure of merit adopted in the speaker's report of the *Herongate's* trials. The coefficient took into account displacement, speed and indicated horse power; whilst the form of a ship remained constant. The unconsidered factor, therefore, was the propeller, and the difference in the figures might therefore be taken as a measure of the respective efficiencies—so far as the formula is trustworthy, and neglecting weather, and the human factor. The latter might fully have accounted for the whole difference—not in the staff of observation, as the speaker had a sufficient number

of practiced assistants, but in the stokehold and engine room complement which belonged to the ship.

The term "slip of the screw" was, of course, used in his report in the conventional sense, and did not allow for the water set in motion by the passage of the ship. It should read "apparent slip." The speaker wished to say a word of regard for Mr. Dickinson, as he feared his remarks about his screw might appear somewhat ungracious. Whatever might be the merits of this invention—and it certainly had merits, as Mr. Walker had very rightly pointed out—the late Mr. Dickinson deserved the thanks of all interested in steamships. He was not working for profit. He was a man of considerable means which he expended freely without hope of return.

The revolutions in the torpedo boat trial bore upon what the author said in his paper as to the friction of the blades. Here there were two propellers of the same diameter, pitch and developed area, running under similar conditions of draft and steam pressure, but one made twenty-four revolutions a minute more than the other. This was due to the different disposition of the blades. Mr. Dickinson's theory was that the forward third of the blade was the efficient part, and this was probably true. But that did not show that one can make a blade all forward third. With regard to the friction of the blades it was well known that the frictional resistance of a surface passing through water was greater at the forward part. So even if Mr. Dickinson could get his blades all forward third there was an augumentation of frictional resistance to take into account, leaving out of consideration his placing two blades close to each other.

Professor Alexander Kennedy thought that the way in which the power had been acounted for would require some revision. He did not propose to speak on the question of the propeller, because Mr. Froude, Mr. Thornycroft and Mr. Barnaby had said all that he could see that there was to say on that subject. He would, however, make some reference to the way in which the load on the engine had been judged. The speaker had made experiments with an engine which was working against a con-

stant resistance. The work was electrical, so that the power necessary to drive it could be accurately gauged. It was generally thought that the power required to drive the engine increased with the load, but this had not been shown to be the case by the experiments in question, as the line on the diagram represented by the power absorbed by the engine was almost parallel with that due to the increase of electrical work. It was, however, far more difficult to get a record of the power with no load on, as the running of the engine would not then be kept so steady. He had concluded, however, that the friction of the engine does not increase with the load in the way commonly supposed.

The author in replying to the discussion said that there was not time to reply to all the important remarks made by Mr. Froude, but so far as he had followed Mr. Froude, their views were in accordance. He was afraid the remarks of many of the speakers had been made chiefly in reference to a particular type of screw propeller; he thought he had sufficiently pointed out in the paper that his reasons for drawing attention to a particular type was because that type happened to have points bearing on the subject of the paper. His primary object was to make comparative tests. The total indicated horse power might be divided into two parts, the internal or friction due to the unloaded engine, and the external, which he had called net horse power; the efficiency of the screws was the ratio of the useful work to the work expended; the work expended consisting of the four quantities which he had summed up as net power. It was clear, he thought, that the power expended in overcoming the friction of the engine should not be included in that quantity; it would, of course, give the combined efficiency of the engine and propeller, but the present problem was quite apart from the efficiency of the engine. The power to overcome the friction of the unloaded engine was obtained by uncoupling the screw shafting during a number of experiments at different revolutions; the steam pressure was practically constant, and rose a little over 8 pounds per square inch; the extra allowance was due to the stern shafting. Mr. Froude considered that some of the frictional quantities were

open to doubt, and on that ground, wondered at the marked equality in the steam pressures due to slip and propulsion. That, the speaker pointed out, was due to the comparative nature of the experiments, he had admitted that some of the quantities were perhaps only approximately correct, but any slight error made would be the same in each case. The element edgeway resistance, to which Mr. Froude had referred, was entirely absent in the experimental propellers, due to the very high slip ratio, and it was partly for that reason that he had chosen a high slip ratio, and thus got rid of that unknown quantity.

Referring to the five constituent parts into which he had divided the indicated horse power, he thought that two only of those factors were open to doubt, the surface friction of the blades and the friction due to working load; the former quantity could be calculated to a very small limit, and taking into account the small amount of power spent in the friction of these propellers, the probable error would come out a very small percentage of the total indicated horse power, and could be left out of account. The friction due to working load was, of course, as Professor Kennedy had pointed out, open at first sight to doubt, and he thought Professor Kennedy had raised a very important point in considering that quantity was constant. It could have been tested by the aid of a thrust dynamometer; the power spent in the propulsion of the vessel could have been obtained by the dynamometer, and was the thrust multiplied by the speed of the vessel, and compared with the power as calculated from the piston pressures. If these two quantities agreed, then the several quantities taken in the analysis were correct. This method was that which Mr. Isherwood followed, and he found that on the surface of the blades there was exactly .45 pound when moving at a velocity of 10 feet per second, and the friction due to working load was exactly $7\frac{1}{2}$ per cent. of the net horse power; and as those experiments were made on a launch about 56 feet long, with engines indicating about the same power as in his own experiments, he felt justified in taking those quantities. He perfectly agreed with Mr. Barnaby's remarks with reference to the paragraph in the paper which stated that the water left the blades at

a greater velocity than the blades themselves, the water flowing through the propeller faster than was accounted for the product of the revolutions and pitch. This had been demonstrated by Mr. Froude in his 1889 paper before the Institution of Naval Architects.

Mr. Dunell had drawn attention to a trial made on one of Mr. Yarrow's torpedo boats. He thought there had been many of what he might term haphazard trials made at one time or another, and with propellers of a similar nature; but they were quite insufficient to settle a delicate point, and scientific conclusions could not be drawn from them. In the first place they were two different propellers, and as the physical surface was not the same, these respective coefficients of blade friction were probably not the same. Mr. Dunnell had been kind enough to supply dimensions of the respective propellers, and from them he should not expect the Dickinson propeller to prove efficient; it demanded a much greater pitch and higher slip ratio in order to balance the loss due to the extra edgeway resistance. Mr. Froude in his remarks had dealt with this point at length.

The late Dr. Froude, in a paper before the Institution of Naval Architects in 1878, noticed that the inclination of the blades and the slip ratio affected the coefficient of friction, that is, the edgeway resistance was much greater at smaller slips than at higher, and it amounts to this, then, the more numerous the blades the greater should be the pitch and slip ratio in order to balance the extra edgeway resistance. Again, Mr. Yarrow's ordinary propeller had blades keyed on, while the Dickinson blades were forged solid on, causing increased edgeway resistance at the roots of the blades. The author also found in his experimental propeller, that unless a certain position was chosen the blades would injuriously affect the water for each other. It was questionable whether Mr. Dickinson had chosen the most efficient position, which was only ascertained by several experiments, and many other minor effects might be mentioned.

He considered one of the most important points in the paper was the one which mentioned that the maximum efficiency of the screw should have been at the working speed. He was sur-

prised this point had not been taken up in the discussion; he would therefore refer to it. Every screw demands a certain slip ratio to work at its maximum efficiency, and it is probable that all ordinary screws attain about the same maximum efficiency, the maximum effect occurring at a particular speed in each case. The problem, the speaker stated, was to design a screw so that its maximum efficiency occurs at the average speed of vessel. Suppose any vessel fitted with any screw, the efficiency curve may be obtained from a series of experiments similar to those in the paper, from which data the design of the screw may be so modified that its maximum effect may occur exactly at any desired speed. At the most it only demands the manufacture of two propellers, the one being the experimental one (which of course may turn out a maximum one), and in many cases only another set of blades.

The author's theoretical knowledge of this subject was, he stated, due to Mr. Froude's paper read before the Institution of Naval Architecs in 1886, and to the experiments of Messrs. Thornycroft and Barnaby. From their experiments it is possible to design a maximum propeller, driving a phantom ship, that is a ship which creates no disturbance in the water, and with an engine having no friction. Mr. Froude has laid down certain constants for applying the results to practice, and which are of course open to error, while doubts are partly eliminated by the above method.

MARINE BOILERS.

[Editorial in "Engineering," February 24 1893.]

The correspondence which has been carried on in our columns during some weeks past indicates the interest taken by engineers in the marine boiler problem. The Admiralty Boiler Committee has now finished its work, and the report has been published, together with the evidence of the various witnesses, in blue-book form. Comment on this, however, is debarred from the fact that it is marked "Confidential," but any good reason for this prohibition we are quite unable to suggest. Of course it is easy enough to formulate bad reasons, and the Admiralty have but themselves to blame if the public attribute unworthy motives for the suppression, supposing, of course, secrecy be ultimately insisted upon. It is, however, impossible to keep these matters entirely secret, and those who take the trouble can get the information. A good number of copies have been struck off, and to divulge secrets which are printed in a book does not prick the average conscience overmuch. No doubt this is what is intended. So long as newspaper criticism and questions in Parliament are avoided, the Admiralty are content. It need not be supposed that the witnesses examined would object to publicity, as there is no more in the evidence of the majority than details of engineering practice which they would communicate in conversation to any of their colleagues or competitors; for, to the credit of engineers, it may be said that professional interest is nearly always stronger than commercial rivalry. In the present instance, as we have often said, the professional officers at the Admiralty need not be ashamed nor fear; but officialism is always timid. Perhaps, however, the report and evidence may be ultimately made public, and British engineers will reap the benefit of a most valuable and interesting mass of facts collected at public expense.

The recent correspondence in our columns was opened by Mr. H. Benbow with a long and able letter which appeared in our issue of October 21 last. The writer's long practical experience in the Navy entitles him to the respectful attention of the engineering profession, and it is satisfactory to us that he echoes many of the sentiments we had formerly expressed. He deprecates the adoption of larger and heavier boilers, which, he says, would be descending the ladder we have set ourselves to climb, and, although difficulties have arisen owing to want of experience, he is quite certain they will be overcome, and trusts this country will not be second in the race. The Admiralty ferrule, he continues, "can at the best only be looked on as a makeshift * * * and it is in the direction of design that we must look for improvement." We have previously described the ferrule as "an expedient," and we think the term well fits the situation. Nevertheless, expediency demanded some device to render available a large number of existing boilers in war ships, and from this point of view the Chatham ferrule may be claimed as eminently successful. For these reasons both Mr. Benbow and "Chief Engineer, R. N.," who appear to join issue on this matter, are in the right. The new ferrule is cheap—we believe the cost is about 8d. apiece—and will last a fairly long time with moderate ratio of fuel combustion, whilst it enables a forced-draft trial to be run with complete success. That is a good record for "a makeshift," but we are sure engineers will agree that boilers ought not to be designed in a manner that makes them dependent on such an expedient.

In our issue of October 28, Mr. James Patterson, of Linlithgow, describes another form of ferrule which he recommends. In this the composition is plumbago ganister and fireclay. "These ferrules extend 8 inches inward and considerably outwards, having a bell mouth very much turned over, thus forming an air space around the contact joint of tube and tube plate, and will withstand any degree of heat that can be put upon them by hot or cold air forced draft, and will last for a considerable time." Before we could accept this device as successful we should like a definite value put upon Mr. Patterson's "considerable time." All

who have had experience in the use of refractory materials subjected to currents of heated gases know how difficult it is to get them to stand for long. The cutting action of the stream of gases and particles carried by them would be extremely trying on the ferrules. We have no experience in such situations of the particular mixture adopted by Mr. Patterson, but in the absence of trial data we should be rather inclined to rely on the "malleable cast" Chatham ferrule, if only because it would be more easily put in position. In a subsequent issue (December 16), we were able, by the courtesy of Messrs. Murray and Patterson, to publish an illustration of the arrangement. By this it will be seen that not only are ferrules placed in the tubes, but they extend some distance outward, and support "a fireproof protective shield," which stands out from the tube plate, thus leaving an air space between. In this way the whole of the tube plate and tube ends are shielded from the impact of the flame, and we can easily believe that the tubes would not leak even under extreme conditions of forced draft so long as the brickwork remains intact. But there is the further serious difficulty from "bird's-nesting" with anything like at forced draft. The whole of the tube plate surface, as heating surface, and the first eight inches of the tubes is lost—at least, so it appears to us according to our reading of the drawings, although another of our correspondents says such is not the case. The question arises whether the advantages claimed could not be attained by more simple means, and we think Mr. Patterson will agree that it is undesirable to build up a masonry structure in the combustion chamber, if it be possible to do without it. It should be stated that Mr. Patterson also adopts what he describes as a "firebrick retort arch," which has the good effect of protecting the junction of the furnace end and the tube plate, and directs the current of heated gases upon the back plate of the combustion chamber, so that they have some of their sting taken out before entering the tubes. Our correspondent does well to call attention to this vulnerable part in the anatomy of the return tube boiler.

If the tube plate has to be protected by a shield of fireproof material, probably the most practical way of attaining this end

yet proposed is that designed by Mr. E. A. Cowper, and illustrated by us on page 795 of our issue of December 23 last. Mr. Cowper's arrangement consists of cheap units easily fixed and readily renewable locally if necessary. As we said, when describing this arrangement, it appears well worth a trial.

Mr. Benbow has evidently studied with advantage Mr. Yarrow's remarkable paper read before the Institution of Naval Architects in 1891, or he has arrived at similar conclusions by independent investigation, for these two engineers are at one upon many details. The former points out that the difference in temperature between the inner and the outer surfaces of the tube plate causes the latter to become convex towards the fire, and on this score he is naturally an advocate of thin tubeplates. On the other hand, Mr. Patterson, in conjunction with Mr. Murray—who joined him in his last letter to us—is "of opinion that thin tubeplates will be short-lived." It would be interesting to hear the reasons our correspondents have for arriving at this conclusion. The facts that have come before our notice point to a directly opposite result; provided, of course, the tube plate be given fair play. For our own part we look on the thickness of the tube plate as the crucial point in the design of boilers to be used with forced draft, and this for the following reason: If a tube plate, say, of $\frac{1}{2}$ inch thickness, can be made to withstand the steam pressure—of course without complication in the matter of stays, &c.—then it is at least on an equal footing with the thicker tube plate, unless Messrs. Murray and Patterson can substantiate their assertion that it will not endure. Now Mr. Yarrow tells us—and his assertion is not a matter of surmise, but of hard, practical experience—that there is no difficulty with his tube plates as regards safety. We know he is directly opposed to stay tubes, believing them to be a fruitful source of evil, and holding that the staying of the ordinary tubes is more than sufficient for the purpose. If that be the case, and our own experience thoroughly endorses Mr. Yarrow's view, the virtues claimed for thin tube plates are all clear gain. There is no *contra* account, and for this reason the primary course of experimental research should be in the direction of thinner tube plates. With regard to ordi-

nary tubes acting as stays, even when loose in their holes, Mr. Yarrow, in a communication which appeared in our issue of September 23, 1892, in reply to a question from the late Dr. Kirk, which was contained in a previous letter to us, makes a significant statement. Dr. Kirk's inquiry was as follows: "Perhaps Mr. Yarrow would give his experience of how far tubes that have become leaky can be relied on to continue to act as stay tubes, as I am sure a full and clear statement from him would go far to restore confidence in every tube its own stay." Mr. Yarrow's reply was to this effect: "As regards the necessity of using stays or stay tubes to resist the strains to which tube plates are subject, for many years we have never adopted them, although, as is well known, our boilers are forced to an abnormal extent, and we have never had the slightest indication of any want of strength, which would certainly have been apparent in some cases, considering that several hundreds of boilers have passed through our hands. This is our experience, not only in cases where no leakage occurred, but also with tubes which in old times leaked to a very serious extent. The above facts, I submit, indicate that, notwithstanding leakage of tubes, they can be relied upon to continue to act as efficient stays if suitably spaced, and assuming the workmanship and materials to be of the highest class. * * * It must be borne in mind that, although when a boiler cools down after trial, the tubes are often found loose in their holes to an extent which would lead to the supposition that there was considerable risk, the same amount of looseness between the tube and plate does not exist when the boiler is under steam; and experience points to the fact that the steaming power of the boiler has stopped long before a dangerous condition of things is reached." We do not forget that Mr. Yarrow's experience is with the loco-marine type of boiler, but his argument bears nevertheless upon the action in a return-tube boiler.

Mr. Durston does not look for salvation in thin tube plates. His contention is that all trouble comes from overheating; that, be tube plates thick or thin, tube ends will leak if the evaporation be rapid, unless circulation of water against the tube plates be unobstructed. We think this is the position taken up by the

engineer-in-chief, if we have interpreted his statements aright. Now it is evident that between Mr. Durston and Mr. Yarrow—and they both have influential supporters—it is a question of degree. Both are right within certain limitations, and the practical problem is where to draw the line. Where does the "hot tube plate man,"—to use Mr. Morcom's expressive term—cease to be reasonable; what are the limitations of the "thin tube plate man?" Our readers will not expect us to dogmatise on this knotty point. To settle it must be a question of experiment, by which alone the proper compromise can be determined. There is, however, the advantage already referred to with which the thin tube plate school start in the contest. If they prove their case, they add nothing to the boiler; on the contrary, they reduce its weight. To provide more efficient circulation it seems almost a necessity something must be added, and that addition is necessary whether the circulation be made more efficient by a greater volume of water being swept over the tube plate, or by reducing the evaporative efficiency of the latter and the tube ends; for circulation of water must always be read in terms of the rate of heat transmitted through a given area of surface. The Admiralty ferrule acts on the latter principle—*i. e.*, reducing the transmission of heat—and it is certainly an addition. Another method advocated is the "plastering of the tube plate," or protecting it by fire tiles, in the manner suggested by Mr. Cowper, and referred to by us above. These methods also entail very substantial additions, not only in themselves, but as a necessitating increase in the size of the boiler, for they mask a considerable area of heating surface which should be very effective. Of course this surface is *not* effective for steam generation if water be not brought in contact with it, and it is for this reason that we look on the increased circulation principle as the more logical course of proceeding of the two. The most obvious manner of increasing circulation is to space the tubes wider; and in connection with this must be considered the area through the tubes for the passage of the heated gases. This is another addition to the boiler in size and weight. In the new designs it has, we believe, resulted in the addition of about 20 per cent. to 25 per cent. to the boilers.

Fig. 1

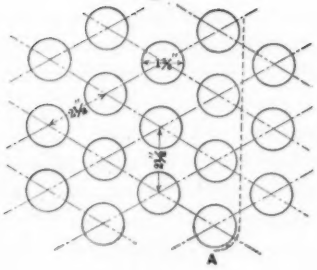
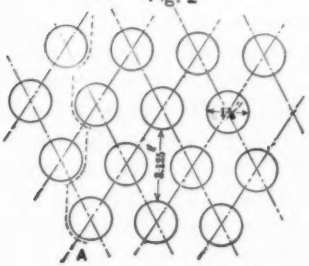


Fig. 2



Mr. F. C. Marshall has found great advantage in one mode of arranging tubes as compared with another, and his experience in this matter is worthy of attention. In two vessels engined by his firm, H. M. SS. *Bellona* and *Barham*, the tubes were placed in zigzag rows arranged as in Fig. 1. In some other vessels built for foreign governments the tubes were also arranged in zigzag rows, but the spacing was different, as shown by the less familiar form in Fig. 2. It will be noticed that in each case the tubes may be considered in lozenge-shaped groups of four, but in the English vessels, Fig. 1, the lozenge is horizontal, whilst with the foreign ships the longer axis of the lozenge is vertical. On the trials of the *Barham* and *Bellona* leakage of the tubes occurred in every case immediately on fires being started, although the air pressure did not at any time exceed 1 inch of water. A large number of tubes had to be removed. With the other boilers there was not the slightest trouble. The *Barham's* boilers had 37 per cent. of the plate occupied by the tubes and 63 per cent. covered by water, the tube ends being $1\frac{5}{8}$ inches in diameter by $2\frac{1}{2}$ inches pitch. In the other design there was 42 per cent. of the plate occupied by tubes and 58 per cent. covered by water, the tubes being $1\frac{3}{4}$ inches in diameter and $2\frac{9}{16}$ inches pitch. Mr. Marshall accounts for the superior performance of the foreign ships' boilers in the following manner*: "Steam leaves the surface on which it is generated in the direction of least resistance, that is, always vertically. The arrangement, therefore, which leaves the greatest amount of free space above the generating surface must be better than one in which obstruction presents itself almost immediately after the steam leaves the surface. In the cases under consideration we have in the vertical plan (Fig. 2) $3\frac{1}{8}$ inches from the top of one tube to the bottom of another; in the horizontal (Fig. 1) we have only $\frac{7}{8}$ inch. It stands to reason that the first must present less resistance to free movement of water and steam, whilst in the other very great obstruction is put in the way of free movement; in fact, there will be no water at all against the tube plate, owing to the steam rising from the top of the tube completely occupying the space. But, in addi-

*Transactions Institution of Naval Architects, vol. xxxii, page 127.

tion to this, we have in both cases the intense heat passing through the plate area between the tubes. If, then, this area is already in both cases filled with steam, as we believe it to be, the steam must be superheated to a greater degree in the smaller space, and the plate and tube ends, therefore, will be more intensely heated than in the boiler where this space is larger. Hence the overheating of both when the space is so limited between tubes in the vertical direction, as in the boilers of the *Barham* and *Bellona*.

Mr. Marshall's reasoning is not quite conclusive, however, or at least not so completely so as he appears to think. It is no doubt true that steam bubbles rise vertically, if possible, when generated, but in both arrangements illustrated they meet with obstruction to vertical movement in the shape of the tubes above, and this to a greater extent in Fig. 2 than in Fig. 1. Although in Fig. 2 there is more room vertically, in Fig. 1 the converging streams of steam bubbles, deflected sideways by the obstruction of the tubes, would not be so likely to interfere with each other in passing. In order to illustrate this matter further, we have added to the diagrams of the tube arrangement the imaginary course of two bubbles of steam supposed to be generated at the summit of the tube immediately below the points A. In Fig. 1 this bubble would, we will say, rise from the spot on the tube where it was generated until it met the obstruction of the tube above. It would then be free to follow a direct course to the surface of the water, so far as any interference from the tubes would be concerned. On the other hand, in Fig. 2, a bubble of steam starting from a corresponding point A would have to bound from tube to tube. Of course our imaginary bubbles would not be able to pursue their calm and natural courses according to their individual volition, but would be jostled, impeded and whirled around by myriads of other bubbles all struggling to the surface. The arrangement shown in Fig. 1 is advocated by Mr. Yarrow, in accordance with locomotive practice, as wounding the tube plate less than the ordinary spacing of the marine boiler. The latter, of course, affords a still more direct and unobstructed passage of the steam bubbles to the sur-

face. Mr. Marshall has, however, actual results on his side, and these cannot be gainsaid, supposing the conditions to be similar in other respects. We believe, however, that the boilers of the English and foreign ships were not alike. Certainly if we make a gain by just turning the lozenges end for end, as Mr. Marshall claims, it is all profit and no loss. It should be added, however, that, as Mr. Marshall states, "Every care was taken to provide for due expansion in all these boilers * * * and, with the exception that in those of Her Majesty's Navy stay tubes were more fully introduced, the boilers should have given equally good results." The tube plates were $\frac{9}{16}$ inch thick, which is a sixteenth in excess of that which Mr. Yarrow laid down, in his paper before referred to, as the maximum thickness.

In a recent issue (see page 760 of our last volume) Mr. Maxim suggests the revival of a device for accelerating the flow of water in the neighborhood of the tube plate. He proposes to throttle the feed so as to gain additional pressure, and then direct it across the surface of the tube plate, thereby inducing a current. His statement that he has found advantage from this arrangement there is no difficulty in accepting; but we are under the impression that his experience has been gained in the circulation through the down-cast pipes of a water-tube boiler. That, of course, is a very different matter to accelerating the flow amongst tubes and over a tube plate. How far the device would go towards curing leaky tube ends in a return-tube boiler running with forced draft is a matter that requires examining closely. An overplus of 50 pounds pressure on the pump side of the check valve is what Mr. Maxim proposes, and the volume of feed, he says, would induce a current in ten times the volume of water in the boiler. That may be true, but it would be at the expense of velocity, and to what extent the advantages of the current so formed would compensate for the additional work thrown on the pump, and the consequent necessity for extra strength in pump and fittings, has to be treated quantitatively and experimentally. Perhaps Mr. Maxim has data on this point. Other devices of a somewhat similar nature—such as a propeller in the boiler delivering a stream of water on the tube plate, and worked by an

engine outside by means of a shaft passing through a stuffing box or pumps taking water from the colder positions and delivering on the tube plate—have been tried, not altogether with success so far as our information goes.

Circulation plates are the next expedient we may notice, and several arrangements have been suggested by our correspondents. The circulation plates demand attention if only from the fact that the first of the cruisers having double-ended single-combustion chamber boilers made successful trials, whilst other vessels of the same type, not having circulation plates, have failed under forced draft. Dr. Kirk's success with the early cruisers, the *Australia* and *Galatea*, has been attributed to circulation plates so arranged that the up-and-down currents of water were divided and not allowed to clash and obstruct each other; in fact, the principle of the Field tube was applied on a large scale. There is this to be said in favor of circulation plates, that they are not expensive, and are easy to fit. There are various arrangements. "W. W.," in our issue of October 21 last, shows how they may be hung on the bottom row of stays, hanging vertically beside the nest of tubes over each furnace, so that they divide each nest from its neighbors throughout the entire length of the tube. Where the evaporation is most rapid over the furnaces and among the tubes, there is an unimpeded upward current, and the steam bubbles are swept to the surface in a continuous procession, whilst the downward flow required to balance this ascending current takes place in the parts where there is solid water, unpierced by tubes. This device our correspondent "W. W." introduced in the year 1864, in two mail steamers, for the purpose of curing the boilers of priming. In this respect the arrangement was successful, although every other device the author could think of had been tried without good result. Mr. Thom, in our issue of November 18, and Mr. Peter Jack, in our issue of December 2, illustrate other methods of arranging circulation plates. "W. W." has no transverse division, but our other two correspondents place a plate a few inches from the tube plate and parallel with the latter, so that the tubes have to pass through it. This device has also been tried elsewhere. Mr. Thom builds up

his complete plate by a number of parts, but this is simply for the convenience of erection. In both cases there are side plates as well, so that the water space through which the first few inches of tube pass is boxed off from the rest of the boiler. The object is the same as that aimed at by the former device, namely, to allow an interrupted upward flow to the water where generation of steam is most rapid; but in the first described arrangement the whole length of the tubes is embraced; in the latter two it is considered desirable to concentrate the action to the vicinity of the tube ends and the tube plate, undoubtedly the crucial part. It is not necessary to point out the differences in detail which distinguish the proposals of Messrs. Thom and Jack, as we have not particulars of results in working, further than to state that the latter arranges his circulation plates so that the steam evaporated by means of the top of the furnace is diverted from the tube plate, and the circulation plates are carried above water-level. Mr. Thom says that his circulating plates are "*now being made* for a vessel worked continually under high pressure with forced draft." We hope the italics are an indication that he will give our readers the benefit of his experience, whether it be favorable or otherwise. It is details of working, not generalities, for which the marine engineer is most grateful at the present time of difficulty and uncertainty.

The method of fixing tubes in their plates has led to some discussion, and there seems a pretty generally unanimous opinion that under modern conditions greater care, at any rate, should be taken in making the joint, even if a new departure should not be originated. Mr. Yarrow dealt with this matter very completely in his paper contributed to the Institution of Naval Architects. He showed the evil effects of a taper mandrel in a parallel hole, and the necessity for a good fit and a clean tube end. Mr. Benbow, in his letter of October 21, lays the greatest stress on this point. He says: "Lastly, most important of all, is the method of fixing the tubes at the combustion chamber end. The old rough-and-ready method of simply expanding the tube ends in place must be discarded as being inadequate to meet the increased strain consequent upon the use of forced draft, and special tubes

as stays be done away with, as they only tend to distort the tube plate, causing the others to leak. Every tube ought to be of equal thickness and form a stay of itself by being properly fitted and secured in place as follows: When the holes in the tube plate are bored they should be faced on the inner side to form a seating of about $\frac{3}{32}$ inch wide and a groove formed around the hole on the outer side of the plate; the ends of the tubes should be upset to admit of a shoulder being turned on them to bear against the seating, then lightly rolled in place, and the ends riveted over and beaded into the groove. This would secure a perfect metallic joint, and the connecting surface between the tube and tube plate would be considerably increased, which is a most important consideration for the free and rapid transfer of heat, and it would prevent the ends of the tubes being burnt away. The other ends of the tubes could be fixed either by expanding and beading in the usual way, or by screwing, the holes in the plate being made large enough to allow the shoulders on the other ends of the tubes to pass through." Mr. Cabena does not approve of this arrangement, and has an alternative proposal of his own, which was illustrated in our issue of November 4 last. The criticism of the latter on Mr. Benbow's letter led to a controversy on this question, which further helped to bring forward the various points raised.

A point to which none of our correspondents have alluded, but one which we ourselves regard as of considerable importance, is that the tubes should be free from rust or scale at the points where they come into contact with the tube plates. We consider that in the case of iron or steel tubes these portions of the external surface should be thorougly cleaned either by grinding or filing prior to the tubes being inserted, and that, moreover, every care should be taken to prevent the parts so cleaned from becoming rusty prior to the tubes being actually fixed.

A matter of some interest that has arisen in the course of this correspondence is whether holes in tube plates expand or contract when the plate is heated. Mr. Turnbull put the matter very cleverly in his letter which appeared in our issue of December 30 last, but he assumes a free tube plate uniformly heated. What

are the actual conditions in service it is difficult to predict, and the readings from experimental data are conflicting. One high authority says that, as a matter of fact, holes do contract; another, equally trustworthy, affirms as a matter of fact they expand. We know they are apt to become oval, but in considering these matters we must bear in mind the relative thickness of tube and tube plate and the consequent relative rapidity with which they heat and cool. That the tubes are in intimate contact with the plate for the distance they pass through the latter is a fact, but, however closely they may be pressed together, there is a break of continuity of metal, and we know that the transmission of heat is largely affected by passage from surface to surface.

There are many other points of interest involved in this momentous phase of marine engineering practice. We do not believe engineers are going to be beaten by the difficulties that have arisen, but that ingenuity and perseverance will again triumph, as they have done in the past in even more difficult situations, and that we shall have marine boilers at once trustworthy, compact, light and economical.

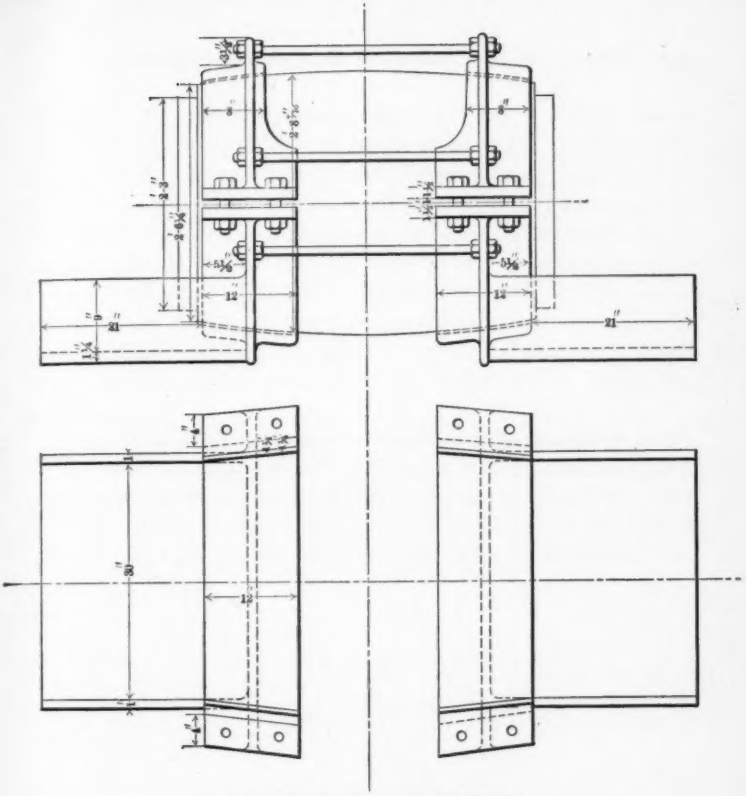
SUPPORTS FOR BORING BARS FOR STERN TUBES AND STRUTS.

By Chief Engineer C. R. Roelker, U. S. Navy.

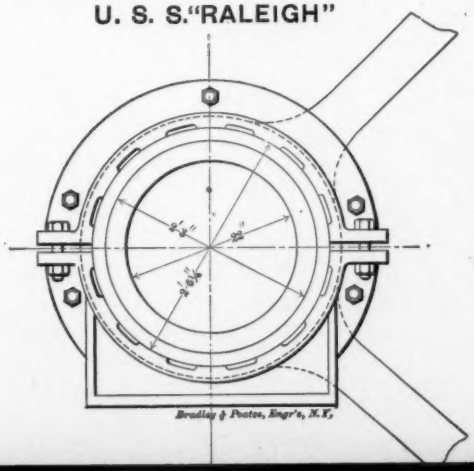
As a supplement to the interesting article on the boring out of the stern tubes and stern struts of the U. S. S. *Cincinnati*, by Passed Assistant Engineer Geo. H. Kearny, U. S. N., which appeared in the February number of the Journal of this Society, I present herewith a drawing of bracket used for supporting the boring bar and gear in boring out the stern struts of the U. S. S. *Raleigh*.

In boring out the stern struts of the U. S. S. *Texas*, it was found to be difficult to construct the wooden trestles supporting the boring bar and gear sufficiently rigid to prevent vibrations. To obtain proper accuracy of workmanship it was, therefore, necessary to take light cuts and run the boring tool at a reduced speed causing loss of time and additional expense. Continual variations in the alignment of the boring bar were likewise observed due, however, to a different cause than the one mentioned by Mr. Kearny. As the tide flowed in and out of the building slip, the lower portions of the wooden trestles were alternately submerged and left bare, and as the boring out took place during the late fall and winter months, when high drying winds alternate with abundant rainstorms in the vicinity of Norfolk, Va., the wood was continually swelling or shrinking, as it became saturated with moisture or dried again, thus throwing the bearings of the boring bar out of line.

To obviate these difficulties the brackets shown in the drawing were designed and constructed for the boring gear of the stern struts of the U. S. S. *Raleigh*. But few remarks will be required in explanation of the drawing. Each bracket is secured



BRACKETS FOR SUPPORTING
BORING GEAR OF SHAFT STRUTS OF
U. S. S. "RALEIGH"



by means of a cast iron clamp, (the lower half of which forms part of the bracket,) to the hub of the stern strut. Facing strips are provided to fit the brackets to a good bearing, and they are tightly bolted together. Then the fore and aft screw braces connecting the two brackets are drawn up hard. That the brackets may be held absolutely rigid on the hub of the strut, they must be made extra heavy, strongly ribbed and braced. The pillow blocks of the boring bar are bolted to the brackets and held in place by side keys bearing against the vertical flanges. These flanges are also used for securing attachments for operating the star feed, etc.

This arrangement worked very satisfactorily and made it possible to run the boring tool, while taking heavy cuts, at as high a rate of speed as the nature of the material of the struts, viz: cast steel with frequent sand holes, would permit. Only one set of brackets was used for the struts on both sides of the vessel.

THE WORTHINGTON MARINE FEED WATER HEATER.

The attention which is now being paid to heating the boiler feed water to nearly the temperature of atmospheric evaporation makes any information relative to improved forms of heaters of great interest.

The Worthington Combined Pump and Heater represents a very carefully worked out apparatus which has proved very successful and satisfactory in steady working on a large scale. The readers of the JOURNAL will, therefore, undoubtedly appreciate the following description:

Fig. 1 shows the external appearance of this heater before it is lagged, and Fig. 2 gives a sectional view of the same and shows its general construction.

Referring to Fig. 2, the exhaust, or live, steam as the case may be, enters the pipe *A* through the valve *B* into the annular chamber and through the perforated plate into the heater. The feed water from the hot well is forced by a pump through pipe *C* to to the valve *D*, and is sprayed through the upper part of the heater, the valve being loaded to open at any desired pressure by means of a spring adjustable by the central spindle. This spray thoroughly mixing with the steam entering through the perforated sleeve in falling to the bottom of the heater, absorbs the heat of the steam and condenses it. The flow of steam into the heater is due to the slight fall of pressure resulting from the condensation of the steam by the feed water.

The steam admission valve is of special construction, which can be opened to admit the necessary amount of steam to the heater, but it closes automatically by gravity in case there is no flow of steam into the heater, as might occur if from any cause the first pump does not deliver water into the heater continuously; and further, this valve being fitted with a dash pot, the closing is gradual; it also prevents the feed water passing into

Fig. 1.

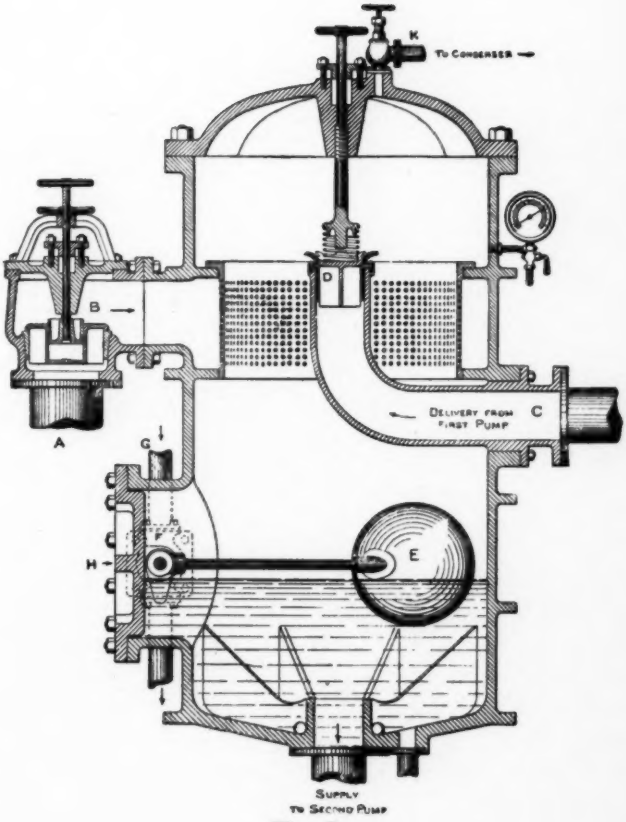

K
To Condenser
D
B
A
Delivery from First Pump
C
G
H
E
Supply to Second Pump

Fig. 2.

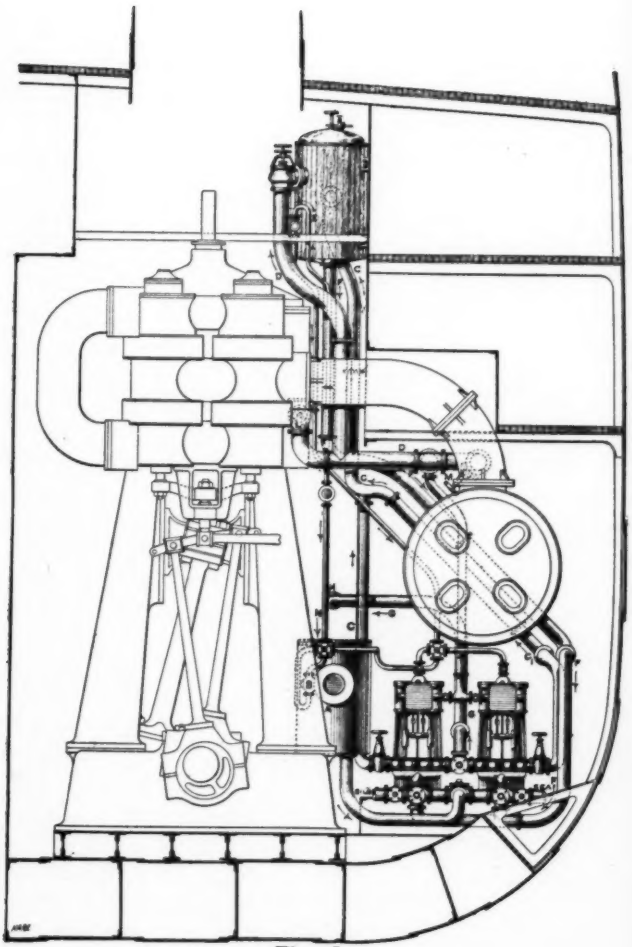


Fig. 3.

the steam pipe should the heater from any cause become filled with water.

The float *E*, shown in the lower part of the heater, is connected to the steam valve *F*, which admits steam to the second feed pump. The working of this pump, which takes the hot feed from the heater and delivers it direct to the boilers, is thus made automatic, depending on the amount of feed water in the heater. The valve *F* is made without any external gland requiring packing, as such packing can never be kept tight for any length of time, and unless great care is exercised in setting up the glands, the valve might stick or permit a leakage of steam in the engine room. In this arrangement should any steam pass the valve it goes into the heater, and is condensed and its heat absorbed by the feed water. The float *E* is without joints or brazing, being made with copper electrically deposited, so there is no possibility of any water leaking into it to prevent its working. The float is tested to an external pressure of 200 pounds per square inch. The float *E* and the head or cover *H* can be removed without in any way disturbing the valve *F* or the pipe connections. The small valve *K*, on the top of the heater, connects it with the condenser, so that any air collecting in the top of the heater is at once drawn off to the condenser. The heater is also fitted with safety valves connecting it with the condenser or low-pressure receiver, as may be preferred, as well as the necessary gauges, etc.

The general arrangement of the Worthington heater with its automatic feed pumps, as fitted on board ship, is shown in Fig. 3.

In the arrangement shown the steam used in the main engines, which is condensed in the surface condenser, is delivered over by the air pumps into the hot well, marked *A*. This delivery being intermittent, it is necessary to regulate the feed pumps so as to take the water from the hot well as it is delivered over by the air pumps in order to insure their complete filling with water. To do this there is in the hot well a float which operates the steam admission valve *B*, and controls the working of one of the pumps, gradually opening the valve as the water rises in the hot well, and shutting it off when the water in the hot well falls be-

low a certain level. The pump in connection with the hot well delivers the water through the pipe C or C^1, as the case may be, into the heater through the spray valve, as already described. The exhaust steam from the various auxiliaries, which is turned into the main exhaust trunk D, is conveyed to the heater where it thoroughly mixes with and imparts its heat to the feed water. The heated feed water, as it accumulates in the heater, raises the float controlling the steam admission valve E of the second pump, and passing from the heater down through the pipe F to this second pump, is forced by it into the main feed supply pipe G.

On the main auxiliary exhaust trunk D is a valve, L, leading to the low-pressure steam-chest casing, and any steam which is not utilized by the heater goes into the low-pressure cylinder, doing useful work; or, if there is not enough exhaust steam from the auxiliaries to sufficiently heat the feed water, steam from the low-pressure steam chest can be passed back through the valve L to make up the deficiency, and a sufficient amount admitted to heat the feed water to any desired temperature. By the valve M the exhaust steam from the auxiliaries can be turned into the condenser when desired.

The arrangement of piping and cocks permits of either pump being used in connection with the hot well or heater, or independently of them, to draw from the sea or for any other desired service.

The feed pumps when working in connection with the heater are perfectly automatic, and practical experience has proved that they require no more attention than pumps worked by the main engine.

Particular attention is called to the fact that no packed valve stems or stuffing boxes are employed in connection with the governing device.

The heater should always be placed at a considerable height above the pumps, and where it is desired to heat the feed water to the highest degree possible this should not be less than 20 feet, in order to prevent any possibility of the pumps becoming filled with vapor, which might occur in pumping hot feed water without a head upon the suction side.

This feed heater, although equally applicable for land use in connection with condensing and non-condensing engines of electric lighting stations and all industrial plants, was especially designed for use on shipboard in connection with triple-expansion and quadruple-expansion marine engines for heating the feed water by utilizing the heat in the exhaust steam from the auxiliaries, and where this is not sufficient, using live steam or steam from the intermediate or low pressure receivers of the main engines.

In the case of marine boilers working under heavy pressures, the practice of feeding with water at high temperatures not only leads to economy of fuel, but also increases the life of the boilers and relieves them from the strain due to expansion and contraction. The common method of heating the feed water by live steam, however, or steam from the intermediate receivers, secures but a small economy, and the main engines are robbed of steam capable of still performing useful work. On the contrary, using exhaust steam, containing as it does nearly the same number of heat units as live steam, gives all the advantages derived from using live steam and utilizes the heat which would otherwise be rejected or lost in the condenser.

It was with the view of securing greater economy by thus utilizing the exhaust steam from the auxiliaries that the Worthington feed heater was especially designed. It is equally suitable for live steam, but such use is only recommended when there is not sufficient exhaust steam for heating the feed water to the desired temperature. On several large steamers, where only exhaust steam from the auxiliaries is used, the adoption of the Worthington heater has raised the temperature of feed water from about 120° Fahr., at which it leaves the hot well, up to 210° Fahr. On steamships employing numerous auxiliary engines considerable economy is thus effected by using the exhaust steam of the auxiliaries in this heater, instead of turning it into the condenser, where its heat is practically lost.

The economy obtained by thus utilizing the exhaust steam from the auxiliaries to heat the feed water is sufficient to justify the use of a combined pump and heater on all classes of steamers.

The economical results thus obtained are much more considerable than would at first sight appear. Take, for example, a triple-expansion engine of 10,000 indicated H.P., and using, say, 15 pounds of steam per indicated H.P. per hour; without any means of heating the feed water, the temperature in the hot well, supplied from the condenser, would probably not be above 120° Fahr., whereas by adopting a feed-water heater the feed water can be raised to 210°, showing an increase of 90 degrees, which corresponds to 13,500,000 heat units saved per hour. Assuming that for each pound of coal burnt 11,000 heat units are utilized, this saving would be about $13,500,000 \div 11,000 = 1,230$ pounds of coal per hour, or, say, 13 tons per day; and if the steamer were working 200 days in the year, a saving of 2,600 tons would be effected.

Worthington feed pumps and heaters have been adopted on some of the largest and most powerful passenger steamers afloat, and wherever introduced the results have been eminently satisfactory. Among the steamships thus fitted are the *Paris* and *New York*, of the International Line. It is a fact that both of these ships made record voyages on the first trip after fitting these heaters early in the past summer, and have several times since broken their own records, showing conclusively that this improvement in the steaming power was, at least, partly due to this apparatus. By the introduction of the heaters on board these steamships the water is now fed into the boilers at an average temperature of about 210 degrees Fahrenheit, instead of as previously, at the ordinary temperature of the hot well; and this has been accomplished entirely by the utilization in the heaters of the exhaust steam from the auxiliary engines. It is quite evident that this increase in the temperature of the feed water must have added materially to the steam making power of the boilers.

[For the foregoing description and for the loan of the cuts, we are indebted to Mr. W. S. Doran, of the Worthington Pump Company.—EDITORS.]

STEAM PIPES.

THEIR MATERIAL, WORKMANSHIP AND ARRANGEMENT.

By W. J. Nowers Brett.

[Read before the Institute of Marine Engineers (England) in November, 1892.]

The subject of steam pipes, which I have the honor to bring before you, has during recent years assumed great importance in the engineering world, and looking at it in the light of some recent experience in explosions and fractures, I think it well worth your earnest consideration and discussion, and hope that ideas will be evolved which, when carried into practice, will give greatly increased confidence to all employed in the marine engine room. Official investigations following accidents to steam pipes have done much to show the defects and the weakness of brazed pipes, and I think I may be excused a digression to point out the great benefit that would accrue if greater prominence was given to these reports in all our technical journals, and in the following paper I propose to quote fully from some to illustrate and emphasize where defects are most likely to occur.

The higher pressures of the present development of the marine engine have caused a certain amount of suspicion to rest upon the brazed-worked copper pipe, and accidents, attended in some cases with lamentable loss of life and injury, seem to confirm engineers in their opinions as to the uncertainty of the ability of these pipes to stand the more severe test now imposed upon them. In the old-fashioned two-cylinder compound engines the brazed copper pipe did its work very well, and taking into consideration the peculiarities of the marine-engine room—vibration, movements of engines and boilers during heavy weather, alterations in the shape of the vessel's hull and the desire not to increase the number of expansion glands—copper seems to be the favorite material, and there is no reason why it should not be used for very high pressures with perfect safety.

Although brazed main steam pipes are looked upon with distrust, the seam of a good copper pipe properly made is as strong as the rest of the material; and for many reasons these pipes are almost exclusively used in marine practice, and consequently this method of manufacturing pipes claims our primary, and I might also add, our earnest attention, because it is likely to be generally used for some time. The greatest danger attending the brazing of a seam is the liability to burn any part of the material, causing the formation of a flaw or crack where it was originally sound, or to partly burn the beveled edges and so make the seam unsafe, and therefore this operation requires all the care of a skillful and trustworthy man. The damage to a pipe by burning is well illustrated in the following report:

"The pipe in question formed one of the branch bends of the main steam pipes of a new Atlantic steamer; its internal diameter was 8 inches; its thickness, .272 of an inch, or No. 2 W.G., and it was made by one of the most eminent firms of coppersmiths in the kingdom with a brazed joint in the usual manner. Before being placed in the ship this pipe was tested by hydraulic pressure to 320 pounds per square inch, or twice the working pressure, on three occasions, with satisfactory results. Nevertheless, when bolted in its place, and while being subjected to a hydraulic test, for the purpose of proving that all the joints were tight, it gave way at a pressure of 310 pounds per square inch, and tore for a length of 20 inches near to the brazing—the material at the point of fracture presenting the appearance of having been partially *cracked during the operation of brazing*. From tensile tests cut from the pipe in the vicinity of the fracture, and also away from the injured part, it was ascertained that the material had a tenacity of 33,000 pounds, showing that the copper was of good quality; and, taking the dimensions of the pipe into consideration, it should have been capable of resisting a pressure of about 2,200 pounds per square inch before giving way; the fact of it failing at a pressure of 310 pounds per square inch is of so alarming a character that * * * it is right to suggest that it is deserving of * * * serious consideration.* This

*Lloyd's Circular.

report emphasizes the necessity of great care on the part of the workman, and also shows how easily copper of good quality loses its virtue.

A great deal depends upon the way the edges are bevelled. It is customary in some places, in the interest of cheapness, to bevel the outside edge and not the inside edge of the seam. It is almost unnecessary for me to say that this method should be deprecated, because it is clear that the thick edge will work into the part overlapping it, and so render the pipe liable to tear or crack in a line parallel with the seam. Both edges should be bevelled, but if made long and thin they are more liable to burn; the bevels of medium length always make the most satisfactory seam. If the bevels are made by hammering, great care should be exercised to prevent the parts adjacent to the seam being reduced in thickness, and furthermore, they should be made to properly overlap one another, and the combined thickness should on no account be less than the uniform thickness of the pipe; but an increased thickness of 25 per cent. or more can be advantageously given as an additional factor of safety. In the best practice the bevels for straight lengths of pipe should be formed on the edges by machine planing, and then, and only then, are true and equal surfaces obtained, and I believe this method is really the cheaper of the two. The fact that properly bevelled edges are of the first inportance for the soundness of a brazed pipe is shown in this report—"The pipe was 7 feet long by 7½ inches diameter, and was made of sheet copper .284 inch thick. The copper was rolled, and the edges bevelled and brazed in the usual manner. It appears that the pipe when first inspected and tested was found unsatisfactory along the line of brazing, the overlapping or joint in some places being open, so that the surveyor could insert his tester about half an inch. The joint was re-brazed at these open places, and the hydraulic test again applied to 320 pounds, which was the test pressure required by the rules (viz: twice the working pressure, which is 160 pounds). The pipe withstood this pressure, but as the surveyor did not feel quite satisfied with the general appearance of the rebrazed parts, he recommended that the pressure be put higher. While this

was being done, one of the re-brazed parts commenced at once to show symptoms of weakness, and at 500 pounds pressure per square inch (which is 340 pounds above the steam pressure) the pipe gave way at that place and opened out for a length of 7 inches. Three samples were cut from the pipe, near to the place where it gave out, one longitudinally, one transverse, and one across the brazed joint. The tensile tests of the two former were 15 tons each, and the latter 12 tons per square inch. The results of these tests show that the copper was of good quality, and had the brazed joint been perfectly made throughout, the bursting pressure of it would be over 2,000 pounds per square inch instead of 500 pounds. The bevelled edges of the copper plate, where the pipe gave way, *were beveled too far back and not sufficiently overlapped, thereby considerably reducing the thickness of the copper at that part, which was torn through the solid metal and not the brazing.*"* The italics are mine, and the words they emphasize point to the necessity for careful workmanship. As with the proverbial strong chain, so with the worked and brazed copper pipe, its strength is that of its weakest place.

Working the beveled edge on sheet copper often determines its quality, and I think it necessary to say that copper is used which after annealing will not stand even that very mild test. Material of this description may possess a comparatively high tensile strength, but it is unsatisfactory to work. As the edge is thinned down by the hammer it has a tendency to crack, and probably to form lamination flaws. The cracks in the overlapping parts of the seam are nullified to a certain extent by the brazing solder, but flaws in the parts adjacent to the seam, although, perhaps, invisible during the test, or closed by planishing the pipe, will develop in use by the vibration of the engines to an extent approaching rupture, and then, under exceptional or unfavorable circumstances, a complete fracture of serious dimensions is the result. This remark, of course, applies to original flaws in sheet copper, due principally to the presence of impurities or blow holes; and another fact necessary for me to mention is, that main steam pipes for high pressures should

*Lloyd's Circular.

not be worked out of sheet copper made from old material which has lost its virtue. I might say it is almost necessary to have a special reliable brand, which would be in the interest of the copper manufacturers as well, because it is imperative that copper as pure as possible should be used, otherwise its attributes as a material for marine-engine steam pipes are gone.

The following particulars from a report* on a steam-pipe explosion, which took place on the steamship *City of Lincoln*, will serve to show the development of flaws in a copper pipe, and furthermore, they will supply useful information, culled from experience. This instance belongs to the category of which the steam pipe of the *Elbe* was a prominent illustration, and includes a number of others more fortunate in the matter of loss of life. The failure on the *City of Lincoln* was in the form of a crack through the solid copper for a length of 8 inches along the edge of the brazed seam. The pipe was $6\frac{1}{2}$ inches internal diameter by about $\frac{1}{4}$ inch in thickness, with a scarph $\frac{5}{8}$ inch in breadth, and was subjected to a working pressure of 160 pounds of steam. The failure, according to the report, was partly due to the presence of water, *but mainly to the fact that the pipe was dangerously weak by reason of old flaws.* The machinery of the vessel was built under the inspection of the surveyors of Lloyd's Registry, and a machinery certificate was granted by that body about nine months before the failure. To ascertain the tensile strength, elongation, &c., of the material forming the pipe, several strips were cut out and carefully prepared for testing; those that were curved were straightened by steady pressure. When pulled in the testing machine only one out of four pieces gave anything like satisfactory results, and even this was deficient in both elongation and contraction of area, while not one of the strips presented the *silky and salmon-pink colored appearance at the fracture which is obtained from sheet copper of good quality.* The bending tests were also unsatisfactory. The results of the tensile tests, with remarks upon the appearance of fracture, are given in the following table:

* Board of Trade.

Dimension of Test Piece.			Ultimate Stress. Per Sq. In. Lbs.	Elongation in 5 in. Per cent.	Contract'n of Area. Per cent.	Remarks, Appearance of Fracture, &c.
Thickness. Inch.	Br'dth. Inch.	Area. Sq. In.				
.255	1.0	.255	24,595	5.0	14.9	Coarse and granular.
.28	1.0	.28	29,590	12.4	20.7	Slightly coarse and granular.
.285	1.0	.285	26,723	6.4	18.6	Coarse and slightly granular.
.26	1.0	.26	32,300	30.0	35.3	Slightly coarse.
.26	1.0	.26	17,046	...	*	Broke through solid copper at outer edge of scarph. Fracture, for ⅔ thickness outside of deep purple color, remaining ⅓ of light pink.
.25	1.0	.25	20,518	2.6	*	Broke through solid copper at outer edge of scarph. Fracture, for ½ thickness from outside, dark brown color, remaining ½ of light pink.
.266	1.0	.265	22,310	2.6	9.4	Coarse.
.27	1.0	.27	24,550	7.0	14.0	Crystalline and slightly granular.

All tensile tests made with strips in normal condition—*i. e.*, cold and not annealed.

* Inappreciable.

The pipe was patched, and when tested to a pressure of 140 pounds per square inch the seam began to leak towards the flange, and when 200 pounds was reached it leaked freely. On examination the seam was found defective for about two-thirds of its length, and the pipe was replaced by one of solid drawn copper. The Engineer Surveyor-in-Chief of the Board of Trade drew a parallel between this pipe and the condition of the exploded steam pipe on the SS. *Elbe*, and in his opinion "the purple appearance of the metal where the defects were located indicated that the cracks either existed in the original sheet of copper or were produced while the pipe was being made." It is also necessary to lay stress upon the evidence of a member of the firm of coppersmiths who bent the pipe to the required shape but did not braze it. He was of opinion, after examining the pipe, that the defective seam and the discoloration through the solid copper was due to a too free use of borax.

It is quite possible that the exploded pipe of the *City of Lincoln* was not properly tested and examined when new, * because the pipe was in such a state that if a hydraulic pressure of double the working steam pressure had been applied the defective condition of the pipe would, very probably, have been made known. This is, I think, a fair and proper test for a copper pipe, and I am of opinion that if the test is carried much above this, it is stressing the pipe to an unnecessary degree. Perhaps the most thorough test for a brazed pipe is the customary one by that eminent shipbuilding firm, Messrs. Denny & Co., viz: the pipes are tested at the coppersmith's to double the working pressure, the brazed seams are examined by hammering along the whole length of the seam and by using a magnifying glass to detect any flaws that may have occurred; and further, the pipes, when jointed up in their places, are again tested from engine stop valve to boiler stop valve to double the working pressure. This is a very substantial test, and should be adopted by all engine builders for new work.

When finishing a brazed-worked pipe it is usual to well planish the seam, a process which does the material no harm if the pipe is properly annealed afterwards; but sometimes the pipe is subjected to the hammer afterwards, when the planishing can be carried to such an extent that the virtue of the material will be damaged by causing it to become hard and brittle—a dangerous quality in a copper steam pipe.

In a number of cases where brazed pipes have failed they have been replaced by those of solid-drawn copper, and in new work they are more than occasionally used. Their manufacture has lately received considerable attention, and by the introduction of the rotative mandrel only a fraction of the power formerly applied is now used, and the material is subjected to considerably less stress. In the best system in practice at Birmingham the copper ingot is pierced with a small hole by a special machine, and the new rotative steel mandrel is then forced through by hydraulic

* There was no special record that this pipe was tested when new, and a member of the firm of Messrs. Doxford & Son—the builders of the machinery but not makers of the pipe—stated in evidence that the pipe was not tested by his firm.

pressure—the other end of the mandrel forming a ram working in a water cylinder, the whole being turned by suitable mechanism. The two-feet ingot with a small hole has now become a "shell" four feet long with a $\frac{3}{5}$ inch bore. It is then passed through the tube-drawing mills, also upon a rotative mandril, and completed.* Solid-drawn copper pipes have, so far, been very successful in marine engine practice, but they are not always free from defects. I have repeatedly seen them cracked or split longitudinally—in some instances for a very considerable distance, varying in sectional thickness to the extent of $\frac{1}{16}$ inch in $\frac{1}{4}$ inch; being below gauge thickness on one side, and exceeding it on the opposite side, thus lacking in uniformity; and the inside of the pipe covered by a multitude of small cracks in all directions and of considerable proportional depth, due probably to overheating during annealing, and I have also detected lamination flaws when the material of the pipe has been experimentally torn through, although at present only in pipes of small diameter but supposed to be of the best quality. In the solid-drawn pipe there is often less chance of detecting a flaw than in the brazed-worked pipe, because by working sheet copper we are generally able to find its faults. A small blow-hole in the ingot of copper will develop into a long and serious flaw—and very often a hidden one, either completely contained in the material or appearing on the surface inside the pipe, where, if it is some distance from the end, it is impossible to see it. These defects show that, though the solid-drawn pipe is a very great advance towards the perfect steam pipe, there is yet further room for improvement in quality and greater care in their manufacture.

In my own opinion the solid-drawn pipe has been excelled by the copper depositing process invented by Messrs. Elmore. I have seen specimens of their manufacture, that appear to be the ideal of copper steam pipes realized. In section they are perfectly true, and of uniform thickness all round, and can be manufactured to any diameter required in practice. They are guaranteed to reach a tensile breaking load of twenty tons per square

* "Machinery for Tube Making, &c., by Mr. W. H. Baraclough, M.I.M.E., read before the Mason College Engineering Society, *vide Engineer*, p. 338, Vol. lxxii."

inch, with proportional elongation, either lengthways or circumferentially. In considering the merits of these pipes it will, perhaps, be better to briefly state the process of manufacture. The tank is made of wood, lined inside with a bituminous composition, and at the bottom of the tank is fixed a perforated copper plate covered with granulated copper. The mandrel (of cast iron, with a thin coating of copper deposited by the ordinary cyanide process) revolves in a solution of sulphate of copper, and between, but not touching, two copper plates to equalize the resistance of the electrolyte, and on completing the connections of the electric current the granulated copper forms the anode* and the mandrel the cathode.† The pipe is deposited upon the mandrel from the solution, but does not adhere to the original copper lining upon the cast iron, because the copper being exposed to the air for a short time produces a coating of ozide, which effectually prevents adherence, and in this way one tube can be deposited over another. The chief feature in this process is the use of a burnisher—a piece of agate—which travels from one end of the pipe to the other, and the pressure of which is varied, together with a difference in the solution, to produce different qualities. The whole process is very slow, as shown by the fact that it takes 168 hours to deposit $\frac{1}{8}$ inch thickness of copper upon a mandrel six inches in diameter. When the pipe has been deposited to the required thickness, the mandrel carrying it is passed through rollers under pressure, which slightly expands and allows it to be drawn off the mandrel.‡ The products of this process leave very little room for criticism from a technical point of view, and the only suggestion I can make is considering the fact that oxide effectually separates the deposited layers—that unless the manufacture is kept up to its proper standard of excellence, if due care and supervision is not exercised, it seems possible that pipes of a scaly nature may be produced. I suppose all engineers have taken a great interest in this most in-

* The positive pole:—The way the current enters the substance through which it passes.

† The negative pole:—The way the current leaves.

‡ *Vide Engineer*, page 178, Vol. lxxi.

genious invention, because it is the means of producing a reliable seamless pipe of first-class quality—a pipe of higher tensile strength than that hitherto obtained from the best sheet copper, and one in which it is impossible to find the ordinary flaws. From a financial aspect the copper deposited pipe bears, apparently, a very favorable comparison with its rivals, its market price being the same as the solid-drawn pipe. But judging from the fact that it takes a very long time to form the deposit, I am inclined to think that if the patents are to be profitably worked, the selling price must be raised.

The bending of copper pipes and the working of copper tee-pieces and pieces with a multeity of branches may be profitably discussed. The easy bends can be dismissed without notice, but where sharp bends are required (and they should not be used if it is at all possible to do without them) they should be made in short lengths of pipe, and if of seamless pipe of greater thickness than the straight length, so that the "back" may not be weaker than the uniform thickness of the straight length. This will make the "saddle" extremely thick in proportion, but it is a defect on the safe side, and will, as the "back" is the smallest arc of the bend and therefore more stressed by movement, allow the bend to easily withstand the working, if not excessive, of the whole range of piping, and therefore it is a source of ample strength when a bend is used to take up the expansion instead of using a gland and stuffing box, although it is an unsatisfactory practice at any time. A brazed worked bend can be made very uniform in thickness throughout by rolling the sheet metal to suit the ultimate shape. A strong pipe can be made by "cramping" the seam, as shown in Fig. 1, which, however, takes longer and requires more care than if scarphed only one way. Tee-pieces or three-way pipes, if made of copper, are worked as shown in Figs. 2, 3and 4. In the two former, seamless pipe can be used to advantage. The latter is composed of several parts worked out of sheet copper and brazed together. The former are undoubtedly the more reliable, Fig. 4 showing a weakness in the center piece. Fig. 2 is the ordinary way branches are made to a pipe, and it and Fig. 3 are the best methods of making copper

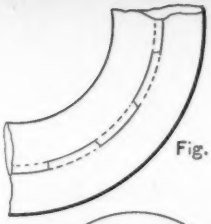


Fig. 1

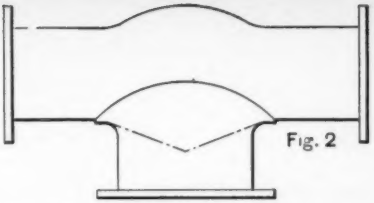


Fig. 2

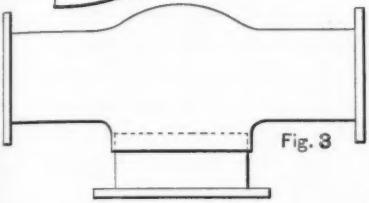


Fig. 3

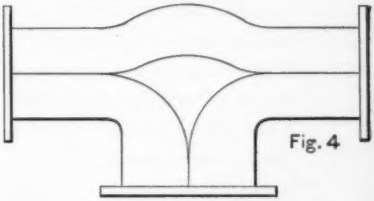


Fig. 4

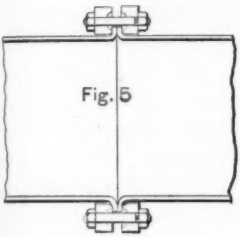


Fig. 5

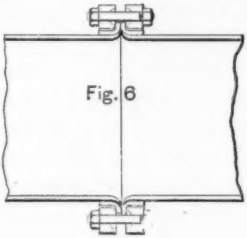


Fig. 6

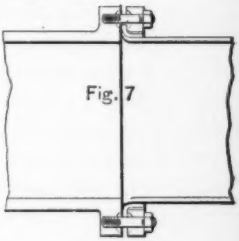


Fig. 7

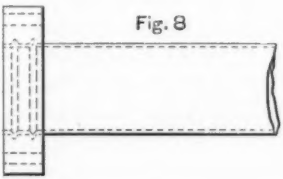


Fig. 8

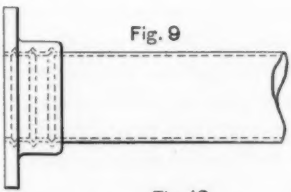


Fig. 9

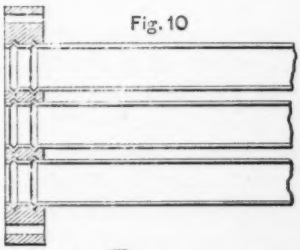


Fig. 10

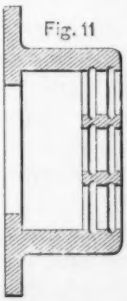


Fig. 11

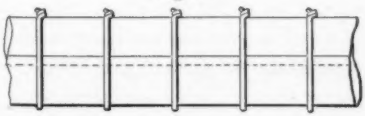


Fig. 12

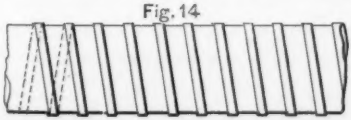


Fig. 14

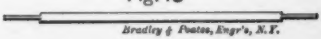


Fig. 13

Bradley & Pontes, Engr's, N.Y.

pieces of three or more ways. But if any doubt is entertained of any copper piece, a gun-metal casting can be substituted, and I should strongly recommend that all branched pieces having weak places, if made of copper, should be made of cast gun metal of ample thickness throughout, and with webs to give the necessary degree of stiffness. It should be much easier to get satisfactory brazing in bends and worked branch pieces than in long straight lengths of pipe, principally because they are more easily fitted together and manipulated over the fire for brazing.

To save the trouble and expense of flanges the socket joint is sometimes used. The socket should not be worked on a seamed pipe, but a seamless sleeve-piece may be used instead. When working a socket-end great care should be taken to keep the stepped part as stout as possible, and it is almost better to reduce one end to fit the other instead of *vice versa*. In any case particular attention must be paid to the running of the brass solder, and it rarely runs the depth of the socket; no joint of this description should be used in the vicinity of a bend.

The brazed flange has been almost as troublesome as the brazed seam, and has contributed its quota to the number of defective steam pipes. The ordinary method of fixing the flange—by beveling the end of the pipe into the counter-sunk side of the flange and then brazing—is sufficient if the flange is a very stout one; but very often the flange is thin, often without a bevel and having very little bearing. Making the flange with a boss gives a good bearing surface for brazing; but there is a medium between having too much and too little. If the boss is a deep one the brazing is more often defective than otherwise, because the brass solder is seldom run completely through, and for this reason it is questionable if the flange with the deep boss is always as strong as its appearance seems to convey. When a sharp bend occurs near a flange, the strength of the brazing of a bossed flange may be advantageously supplemented by a few copper rivets.

The only method differing from the ordinary practice that has made such headway in marine work is the flange patented by Mr. Pope, and shown in Figs. 5, 6 and 7. The ends of the pipes

are belled out to form the jointing faces. If seamless pipes are used the flanges can be formed as shown in Fig. 5, but—and I beg to differ—Mr. Pope prefers the form shown in Fig. 6, in which the copper flange-pieces are worked separately and then brazed on the ends of the pipe. The jointing rings XX can be made of cast steel or iron, or malleable steel or iron, but cast steel is preferable. When the rings are made in one piece, they have to be strung on to the copper pipe before both ends are flanged out, but they can be made in halves, half checked at the meeting ends. I have seen this method of flanging and jointing used with a working pressure of 250 pounds per square inch in marine practice.

The idea patented by Mr. de Ferranti was the outcome of the explosion at Deptford, and although ingenious and applicable to stationary engines with a good foundation, is not, in my humble opinion, quite suitable in its present form for marine use. I have been informed that it has given complete satisfaction where applied to the boilers and engines of the Electric Supply Corporation, but if applied to steamships it will necessitate a greater safeguard, owing to the movements of the structure. The flanges are secured to the pipe by corrugations as shown in Fig. 8, but for marine fitting I would suggest that it be slightly altered, as shown in Fig. 9, the grooves made much deeper and the number increased, and it might give additional satisfaction to bore the flange slightly taper to prevent any possibility, however remote, of the pipes being drawn out. The tests conducted by Mr. de Ferranti showed that it took a hydraulic pressure of many times the working pressure to destroy the fixture, and then, after almost reaching the bursting pressure, *one* of the corrugations drew out of the groove. The higher the pressure the tighter the pipe was in the flange, but it must be remembered the pressure, being hydraulic, was steady. In marine practice, and especially near a bend, the vibration of the engines when steaming and the movements due to heavy weather, in addition to the pulsation of the steam (if near the engines) would probably have a pernicious effect unless something in the way of the preceding modification was introduced. However, the idea is a great stride

in the right direction, for brazed flanges not being always reliable, it is only natural that the thoughts of the engineer should be directed to a different channel—to do without brazing altogether.

To Mr. de Ferranti is also due the credit of practically introducing the system of bunching together a number of small solid-drawn or seamless pipes to form the required area of a large steam pipe, and by this means minimizing the danger and destructiveness of an explosion. The multiple of the pipes can be anything, and, as shown in Fig. 10, the flanges can be made to suit. The pipes are secured in the flanges by corrugations of definite pitch and depth, and where the range of piping ends in a stop valve, dome or casing, the flange shown in Fig. 11 is used. These multiple pipes are easily applied to bends, but where a tee-junction occurs a cast-metal piece is necessary. I do not think Mr. de Ferranti's system requires further explanation, and it is only necessary for me to say that for land-engine practice its merits have been unquestionably proved, and its adaptation to marine-engine conditions seems perfectly feasible.

Various devices have been resorted to to secure a comparative amount of safety in case of an explosion of a brazed seamless copper steam pipe in the marine-engine room and stokehold, which otherwise present few facilities for the hurried escape of those employed there. These devices have been adopted to reduce or prevent the escape of steam. Of the former perhaps one of the best is by binding the pipes with copper or steel wire, which does not reduce the flexibility of the pipe, and would, if the pipe gave way, prevent the fracture from being of serious dimensions, and would, therefore, allow time to close the stop valve. It was this, I presume, that suggested to Mr. Peter Denny the idea of strengthening pipes by encircling them with hoops placed at suitable distances apart along their length and binding them as tightly as possible. The strengthening hoops are made of wire or rods of malleable iron, steel or bronze, or other suitable metal (see patent specification), and are closed and tightened upon the pipe by twisting the ends together, as shown in Fig. 12. With all respects to Mr. Denny's invention (and I

am indebted to his kindness for some necessary information in this paper), I would suggest that the hoops be made as shown in Fig. 13, which, giving greater bearing surface, would be less likely to damage the pipe by "grooving." Great care should be taken in fixing the hoops, as not only can the iron be strained, but the pipe as well, and more especially so if the hoops are made red hot for twisting. It occurs to me that a more commendable method of strengthening a brazed pipe is by winding round it a flat spiral copper strip at a given pitch (Fig. 14), and either brazing it to the pipe along the line of the longitudinal seam when the seam is brazed, or separately, or tinning and soldering on the spiral in convenient lengths, and this, I think, would give more satisfactory results than hoops or wire winding. Means have been taken to wholly prevent the escape of steam by placing the copper pipes entirely within a steel or iron casing —a very expensive arrangement when a long range of piping is dealt with.

The failure of a copper pipe often repeated naturally suggested to some engineers the desirability of dispensing with copper altogether as the material, and using another. Wrought iron and cast iron have been occasionally used since the earliest days of marine engineering, but we have few expressed opinions upon their behavior. Recently wrought iron and steel lap-welded pipes have been fitted to high pressure marine engines with apparently satisfactory results, the greatest possible care having been taken to provide an ample number of expansion glands to remove all probable strains due to structural conditions. Manufacturers are now directing their attention to improvements in the production of solid-drawn steel tubes, with a view to supersede copper, and having this before us, and the fact that solid-drawn or lap-welded steel or iron pipes are considered likely to be eventually universally used, it will be well to note the working conditions of a steel or iron pipe. In the first place, we have a formidable enemy to encounter in the corrosion of the material and the formation of oxide, and its ultimate passage into the valve chambers, cylinders and glands. Secondly, we have extreme rigidity making it necessary to provide frequent expansion glands; and, thirdly,

they are just as likely to have defective welds as defective seams in the brazed pipe, and the solid-drawn pipes, either steel, iron or copper, will probably continue to have their defects and flaws, even if the steel and iron pipes be made many times the thickness of the present copper ones.

Some time ago an interesting paper dealing with the use of iron main pipes was read before a kindred Institution.* The author advocated their use in lieu of copper ones, and in the course of his paper mentioned his experience of an iron lap-welded pipe fitted on board the SS. *Claremont.* The pipe was 5 inches external diameter, and fitted with screwed and brazed flanges, and the steam speed was low, the pipe being large in comparison with ordinary practice. The original thickness of the pipe was .375 of an inch, and after $8\frac{1}{2}$ years' continuous service the thickness was practically the same as when new. The interior showed no signs of pitting or corrosion. It was covered with a thin coat of black oxide nowhere exceeding $\frac{1}{32}$ inch in thickness, and where the deposit was thickest it was curiously striated by the action of the steam. On the scale being removed the original bloom on the surface of the metal was exposed, and externally the pipe, which was not covered, suffered slightly from corrosion. The pipe, it must be admitted, possessed the qualifications expected, but I cannot help thinking the conditions under which it was tried must have been very favorable.

The priming of the boilers would have a severe excoriating action upon the internal portion of iron steam pipes, and in exceptional cases the severe scouring would quickly shorten their lives, for even the great resistance of copper to this action is slowly overcome by it.

The general arrangement of main steam pipes claims attention. An ordinary range of piping should be supported at certain distances throughout its length, and must not be rigidly fixed in any way, as in the case of a watertight bulkhead, but should be left free to take up the movements of engines and boilers. An interesting lesson may be found in the failure of a brazed copper pipe on board the SS. *Ragusa.* This pipe connected the two

* North East Coast Institution of Engineers and Shipbuilders

stop valves on the main boilers. It was 6 inches internal diameter, about $\frac{1}{8}$ inch in thickness, and had a brazed seam running along each side and the ends were flanged in the usual manner. It had always given trouble, and during heavy weather the pipe burst and the fracture opened and closed with the rolling of the ship. Mr. Traill, the Engineer Surveyor-in-Chief of the Board of Trade, in his observation, says: "The pipe * * * was obviously too rigid, and being the only connection between the upper parts of the boilers, it is not surprising that it has given so much trouble. It is well known that boilers on board ship frequently move considerably during heavy weather, and it is the duty of those who are responsible for the fitting of the steam pipes to see that they take no part in controlling that movement." The arrangement for taking up the movements and expansions of pipes are important, and, as such, deserve due consideration. In all cases where the design of the pipe is made to allow for its movements, I think the material under this condition and the action of vibration and temperature is unduly stressed and liable to rapidly deteriorate in quality, and probably to crack; but if the safest form of gland-and-socket expansion joint with safety collar to prevent drawing out is properly placed and fitted, the resulting evil of this triple action is very small.

Sharp bends must be avoided if at all possible, and other arrangements substituted, and in the general design as few ordinary bends as possible introduced. All ranges of piping must be efficiently drained, and drain pipes fitted wherever condensed steam or water from priming is likely to collect. Water in a steam pipe acts like a hammer, and all pipes are subject to its stress more or less. It has been the primary cause of several explosions—some quite recently. The steam pipe of the SS. *Rohilla* exploded "owing to the stresses brought on it by the moving water within," and the internal surface of this pipe must have been frequently subjected to the scouring action of water. Water-hammer action is a grave danger, and might be the cause of a perfectly sound pipe failing. As an instance of the force exerted by a water hammer blow, it may be mentioned that the

main engine stop-valve cover—a very strong one—was severely cracked in a radial direction by the above explosion. At the inquiry into this accident a surveyor suggested that special fittings should be provided so that the drained water could be seen, and thus the amount of water formed and the efficiency of the drain ascertained.

In the fitting of steam pipes, I must strongly denounce the practice of drawing pipes up to their places with bolts if they do not fit. Where this practice is followed I should be inclined to think that the fixing of the engines and boilers themselves was not satisfactory, and the resulting vibration acting upon a pipe or pipes held in a strained position is sufficient to damage even the best brazed seam or flange. More attention should be paid to the fixing of engines and boilers and to the balancing of the working parts of engines, and I am glad to see the attention of one eminent experimentalist at least diverted in this direction. Vibration is not only a source of discomfort but a danger, a deteriorating action upon the whole structure, so we cannot expect steam pipes to remain unaffected by it.

An important point in favor of copper as a material for steam pipes is that it can, from its nature, take up the greatest number of small unknown strains that must occur in ranges of pipes, better than any other material, and thereby give less trouble with the joints. Its most serious defect is its loss of strength with every increment of temperature, but the day is distant when we shall have to increase the margin of safety for compensation. So far copper pipes have rarely, if ever, burst in the material apart from the seam or flange, but I would suggest a greater margin of safety than that now generally used, and this is becoming more recognized in new work. Regarding brazing, I reiterate the remark that, if properly done, it will be equal to the strength of the copper itself, and as it is likely to be used, although so often denounced, I hope, as a safeguard, to see the introduction of a brass solder of great strength and comparatively low running temperature. By this means we would get a more reliable brazed pipe, because the possibility of burning the material would be more remote. Testing by hydraulic pressure

is an unfair test by itself, and should not be carried above twice the working pressure; but the seam should, in every case, be tested by the light use of a hammer and the magnifying glass while under this pressure, which I mentioned before as being used by Messrs. Denny, and in addition to this the inside could be examined afterwards by a specially-arranged light and reflector. The fire over which the pipe is brazed should be freed from impurities, and by this precaution a better contact with the solder would be obtained and the discoloration of the joint avoided. The rules of the Board of Trade, that surveyors may test copper steam pipes in their places at any time they consider necessary, should, in my opinion, be carried further. They should be taken down once in a given period, and annealed and tested by hydraulic pressure and as previously described. If they remain in their places for some years without annealing they become brittle, probably from the vibration of the engines, and they are then in a dangerous condition; and, further, the passage of old copper pipes through the fire would cleanse them, and would, if there were any open flaws or cracks, open them out and make them visible. This is, although a more expensive method, the only way to efficiently examine and test old copper steam pipes.

THE CONTRACT TRIAL OF THE U. S. S. *BANCROFT*.

By Passed Assistant Engineer Robert S. Griffin,
U. S. Navy.

The *Bancroft* was built by the Samuel L. Moore & Sons Co., Elizabeth, New Jersey, from designs furnished by the Navy Department, the contract price for her construction being $250,000.

She is barkentine rigged to top-gallant sails, without bowsprit, and has a poop and top-gallant forecastle and a bridge with pilot house just abaft the forecastle. In outward appearance her hull resembles somewhat that of the *Baltimore* or *Philadelphia*, and may be considered a small edition of one of those ships. She has a ram bow, with a torpedo tube discharging above the water line, and a broadside torpedo port on the starboard side of the berth deck. A light water-tight or protective deck extends from stem to stern, and a coffer-dam, intended for cellulose, from the forward to the after magazine, including the engine and boiler rooms. The water-tight sub-division is similar to that of the larger ships, except that there is not a double bottom.

Much of the space is taken up with accommodations for the Cadets, but in order to accommodate about forty of them, it would be necessary to send about fifty or sixty of the crew out of the ship, and then it would be practically impossible to stow all the hammocks during the day time, as three of the Cadets' bath rooms are now utilized for this purpose.

The armament consists of four 4-inch R.F. rifles, two 6-pounder Hotchkiss, two 3-pounder Driggs-Schrœder, one 1-pounder Hotchkiss, one 37 mm. Hotchkiss, and a Gatling gun.

HULL.

Material	Mild steel.
Where built	S. L. Moore & Sons Co.
When built	1890–1893.
Length between perpendiculars	180 feet.
Length on L.W.L.	187 feet 6 inches.
Length over all (including rudder)	189 feet 6 inches.
Beam, extreme	32 feet.
Beam at L.W.L.	32 feet.
Ratio length (B.P.) to beam	5.63
Depth in hold from top of main deck beams to top of floors	19 feet 5½ inches.
Draught, (moulded) forward. for L.W.L.	11 feet.
Draught, (moulded) aft. for L.W.L.	12 feet.
Draught, (moulded) mean. for L.W.L.	11 feet 6 inches.
Depth of bar keel below skin	8 inches.
Displacement at load draught	839 tons.
Displacement per inch at load draught	9.52 tons.
Area of immersed midship section	279.36 square feet.
Area of L.W.L. plane	4,002.29 square feet.
Center of gravity of L.W.L. plane aft of midship section (frame No. 51)	2.31 feet.
Transverse metacenter above center of gravity	1.91 feet.
Longitudinal metacenter above center of gravity	212.88 feet.
Coefficient of fineness on extreme dimensions	.425
Coefficient of fineness on midship section	.758
Coefficient of fineness of L.W.L.	.667
Wetted surface (by girths)	6,456 square feet.

MACHINERY.

The engines and boilers are from the Navy Department's design, and comprise two vertical, inverted, direct-acting triple-expansion engines, and two horizontal fire-tubular boilers of the straight-way or gunboat type, together with the necessary auxiliaries. Both engines are in the same water-tight compartment, and the tops of the cylinders are below the water-tight deck. The boilers are in the compartment immediately forward of the engine room, and are fired from the after end. There is one condenser common to both engines, with independent air and circulating pumps. The forced draft is on the closed ash-pit or Kafer system.

Each engine has three unjacketed cylinders—the high-pressure, which has a working liner, forward, and the low aft, the axes of each set being in a plane parallel to the center line of the keel.

They are supported from the bed plates on wrought-steel columns trussed by tie rods; the bed plates are of cast steel in three sections. There is one piston valve for the high and two for the intermediate, and a double-ported slide for the low-pressure cylinder, all worked by Stephenson link motion. The piston valves are solid and take steam at the ends. There is a hydraulic reversing engine with hand pump attached for each main engine. The reversing arms are slotted for a sliding block for alteration of cut-off, which is variable between the following points: H.P., from .52 to .68; I.P., from .52 to .70; L.P., from .52 to .71 of the stroke.

The pistons, cross-heads, eccentrics, and cylinder and valve chest covers are of cast steel, and the piston-rods, connecting rods, valve stems and valve gear of wrought steel.

The following are some of the principal data:

Cylinders, number for each engine		3.
Cylinders, diameter of H.P		13½ inches.
Cylinders, diameter of I.P		21 inches.
Cylinders, diameter of L.P		31 inches.
Stroke of pistons		20 inches.
Valves, diameter of H.P		6 inches.
Valves, diameter of I.P. (2)		6 inches.
Valves, diameter of L.P. balance piston		3¾ inches.
Valves, effective area of L.P. balance piston		5.1 square inches.
Valves, length of L.P		2 feet 0¼ inches.
Valves, width of L.P		2 feet 3 inches.
Area H.P. steam ports, maximum opening	Top	10.60 square inches.
	Bottom	12.76 square inches.
Area H.P. exhaust ports, maximum opening	Top	26.70 square inches.
	Bottom	22.38 square inches.
Area I.P. steam ports, maximum opening	Top	23.56 square inches.
	Bottom	29.84 square inches.
Area I.P. exhaust ports, maximum opening.	Top	53.40 square inches.
	Bottom	44.77 square inches.
Area L.P. steam ports, maximum opening	Top	49.50 square inches.
	Bottom	53.25 square inches.
Area L.P. exhaust ports, maximum opening	Top	92.25 square inches.
	Bottom	85.50 square inches.
Valve stems, H.P., diameter		1¼ inches.
Valve stems, H.P., diameter through valve		1 inch.
Valve stems, I.P., (2) diameter		1¼ inches.
Valve stems, I.P., (2) diameter through valve		1 inch.
Valve stems, L.P., diameter		1¾ inches.

Valve stems, L.P., diameter through valve ..... 1⅜ inches.
Main steam pipe (4½ inches diameter), area cross-section ..... 15.90 square inches.
Exhaust pipe to I.P. (5¾ inches diameter)..... 25.96 square inches.
Exhaust pipe to L.P. (7¾ inches diameter)..... 47.17 square inches.
Exhaust pipe to condenser (9¾ inches diameter)..... 74.66 square inches.
Volume swept by H.P. piston per stroke ..... 1.628 cubic feet.
Volume swept by I.P. piston per stroke..... 3.980 cubic feet.
Volume swept by L.P. piston per stroke..... 8.707 cubic feet.
Ratio of net area of H.P. to I.P. piston..... 2.42.
Ratio of net area of I.P. to L.P. piston. ..... 2.177.
Ratio of net area of H.P. to L.P. piston..... 5.267.
Clearance in H.P. cylinder, per cent..... 7.9.
Clearance in I.P. cylinder, per cent..... 10.4.
Clearance in L.P. cylinder, per cent..... 7.1.
Piston rods, diameter..... 2½ inches.
Piston rods, length from piston to cross-head..... 2 feet 4½ inches.
Connecting rods, length center to center..... 40 inches.
Connecting rods, diameter of upper end..... 2½ inches.
Connecting rods, diameter of lower end..... 4 inches.
Connecting rods, thickness (sides are faced)..... 2½ inches.
Connecting rods, cross-head bolts (4)..... 1½ inches.
Connecting rods, crank-pin bolts (2)..... 1⅝ inches.
Cross-heads, surface (9 × 8)..... 72 square inches.
Cross-heads, pins, diameter..... 3¼ inches.
Reversing gear, steam cylinder, diameter..... 6 inches.
Reversing gear, oil cylinder, diameter..... 3½ inches.
Reversing gear, stroke..... 10⅜ inches.
Reversing gear, piston rod, diameter..... 1¼ inches.
Reversing gear, hand-pump plunger, diameter..... ⅞ inch.
Reversing gear, hand-pump plunger, stroke..... 4½ inches.

Shafting.—The shafting is of steel throughout. The crank shafts are each in three interchangeable sections, connected to each other and to the thrust shaft by six steel bolts. The after end of the thrust shaft is enlarged to meet a coupling keyed on the shaft through the stern tube, in the forward end of which shaft there is a central stud, with a nut bearing against the collar of the coupling, to take the thrust in backing, the end of the thrust shaft being swelled to admit the nut. The stern shaft is cased with composition for its entire length, and the propeller shaft only where it passes through the strut.

Crank-shafts, diameter ..... 6 inches.
Crank-shafts, axial hole ..... 3 inches.
Crank-shafts, coupling disc, diameter ..... 12 inches.
Crank-shafts, coupling disc, thickness ..... 1½ inches.
Crank-shafts, coupling bolts (6), diameter ..... 1⅜ inches.
Crank-shafts, journals, diameter ..... 6 inches.
Crank-shafts, journals, length ..... 5½ inches.
Crank-shafts, length of each section ..... 2 feet 7½ inches.
Crank-pins, diameter ..... 6 inches.
Crank-pins, length ..... 6 inches.
Crank pins, axial hole ..... 3 inches.
Crank-webs, width ..... 7½ inches.
Crank-webs, thickness ..... 4 inches.
Thrust shaft, diameter ..... 5¾ inches.
Thrust shaft, diameter of after flange ..... 14¾ inches.
Thrust shaft, axial hole ..... 2⅞ inches.
Thrust shaft, collars, number each shaft ..... 7.
Thrust shaft, collars, diameter ..... 9 inches.
Thrust shaft, collars, thickness ..... 1½ inches.
Thrust shaft, collars, distance between ..... 1⅝ inches.
Thrust shaft, surface, total for both engines ..... 527.1 square inches.
Thrust shaft, length ..... 10 feet 5 inches.
Stern shaft, diameter ..... 6 inches.
Stern shaft, axial hole ..... 3 inches.
Stern shaft, diameter of keyed coupling ..... 9 inches.
Stern shaft, length of coupling on shaft ..... 5¾ inches.
Stern shaft, diameter of coupling flange ..... 14¾ inches.
Stern shaft, diameter of central stud ..... 2⅞ inches.
Stern shaft, diameter of coupling bolts ..... 1⅜ inches.
Stern shaft, length ..... 29 feet 4 inches.
Propeller shaft, diameter ..... 6 inches.
Propeller shaft, axial hole ..... 3 inches.
Propeller shaft, diameter of coupling ..... 12 inches.
Propeller shaft, diameter of coupling bolts (6) ..... 1⅜ inches.
Propeller shaft, length ..... 21 feet 2½ inches.

Propellers.—The propellers are of manganese bronze, three bladed. The blades are adjustable and are each secured to the boss by six 1⅜-inch Tobin bronze bolts; their general shape is elliptical, and they are bent back through an angle of about 24°. The boss is secured to the shaft by a taper key and a composition nut, and the end is covered with a composition cap.

Number of blades ..... 3.
Diameter ..... 7 feet 0 inches.
Diameter of hub ..... 1 foot 10 inches.

Pitch as set (adjustable from 6 feet 9 inches to 8 feet 3 inches)	7 feet 9 inches.
Greatest width of blade (2 feet from axis)	27½ inches.
Helicoidal area of each screw	15 square feet.
Projected area	10.66 square feet.
Disc area	35.845 square feet.
Pitch ÷ diameter	.903
Helicoidal area ÷ disc area	.418
Projected area ÷ disc area	.297
Disc area of both screws ÷ I.M.S.	.256
Immersion of center at mean draught (11 feet 6 inches)	7 feet 6½ inches.
Center above base line	3 feet 11½ inches.
Center from center line of vessel	4 feet 6 inches.

Condensers.—The main condenser shell is cylindrical, of composition, in one piece, and is placed immediately abaft and between the two main engines, with the tubes running athwartship and parallel to the axis of the condenser. The circulating water enters on the starboard side, at the bottom, and returns in the upper half, the inlet chest being divided for this purpose. There are straightway valves connecting the condenser with each exhaust pipe, so that in the event of accident to either engine the vacuum may not be impaired by leakage. The auxiliary exhaust is also connected with this condenser.

Diameter of shell	4 feet 0 inch.
Length over heads	7 feet 8¼ inches.
Tubes, diameter outside	⅝ inch.
Tubes, length between heads	6 feet 3½ inches.
Tubes, number	1,772.
Condensing surface	1,824 square feet.

There is, besides, an auxiliary condenser of the Davidson type, with combined air and circulating pump, intended for use when not steaming. It is placed on the starboard side, just outboard of the main engine. Its principal dimensions are:

Diameter	17 inches.
Length	6 feet 5 inches.
Tubes, outside diameter	⅝ inch.
Tubes, length between heads	5 feet 0 inch.
Tubes, number	195.
Condensing surface	160 square feet.

Combined pump:

Steam cylinder	5½ inches.
Water cylinder (circulating)	6 inches.
Air pump cylinder	7 inches.
Stroke, common	8 inches.

Air-Pumps.—The main air-pump is a Davidson pumping engine, consisting of a vertical inverted steam cylinder and two vertical air-pump cylinders, whose buckets are operated by means of a beam worked from the crosshead of the steam cylinder. There is a connection from the channel way of this pump by means of which the water of condensation can be pumped from the condenser (when used for auxiliary purposes) by either of the feed-pumps.

Diameter of steam cylinder	8 inches.
Diameter of each pump cylinder (two)	14 inches.
Stroke, common	12 inches.
L.P. piston per stroke ÷ air-pump, per stroke of steam piston	16.483

Circulating Pump.—The circulating pump is of the centrifugal type, driven by a vertical steam engine.

Diameter of engine cylinder (one)	6 inches.
Stroke	5 inches.
Diameter of pump runner	22 inches.
Width of fans	at hub, 4½ inches; at tip, 2 inches.
Diameter of inlet nozzle	8 inches.
Diameter of outlet nozzle	7 inches.

This pump is provided with a suction pipe from the bilge, and a special discharge to the outboard delivery pipe, so that if used for freeing the ship, the water need not be pumped through the condenser, there being a straightway valve in the delivery to condenser for this purpose.

Turning engine.—A small turning engine is placed in the passage between the engines. It is a double engine, 3½ and 3½ by 4 inches stroke, and is provided with disconnecting gear, so that the engines may be turned by hand with a bar.

Feed pumps.—The main and auxiliary feed pumps, one of which is in the fire room and the other in the engine room, are vertical, and of the Davidson type.

Auxiliary pumps.—Besides the feed pumps, there is a fire and bilge pump, a water service pump, a distiller circulating pump, and an evaporator feed and tank pump, all of the Davidson type. The auxiliary feed, the fire and bilge, the water service, and the distiller circulating pumps are connected with the fire main. The pipes for flushing closets and bath rooms connect with the fire main, so that any of these pumps may be used for flushing when the distiller is not in operation.

The following are the sizes of these pumps:

	Steam Cylinder.	Water Cylinder.	Stroke.
Main feed	9	5½	10
Auxiliary feed	9	5½	10
Water service	9	5½	10
Fire and bilge	12¼	7¾	12
Distiller circulating	5½	5	8
Evaporator feed and tank	3	1⅜ / 1½ tank.	4

Distilling apparatus.—The distilling plant consists of a Baird evaporator (No. 3, Type C) and distiller, having a rated capacity of 2,000 gallons of potable water a day. The evaporator shell is a vertical cylinder, the tubes being U-shaped, disposed in horizontal planes. There is a connection to main condenser for making up loss of feed water. The distiller circulating pump discharges into the fire main and has a by-pass, so that, when not distilling, the water may be delivered into fire main without going through distiller. The evaporator feed and tank pump takes its feed from the distiller discharge water, and the tank part takes its suction from the filter through a water meter and discharges into ship's tanks and into firemen's wash-room tank.

Revolution counters and tell-tales.—The counters are rotary and run by link belts from small horizontal shafts which receive their motion, through gearing on a vertical shaft, from a worm on the end of the crank-shaft. The motion of the small shafts is reduced 10 to 1, so that in using the ordinary counter it was necessary to put an additional wheel counting units on the shafts. With this arrangement, when the engines are reversed, revolutions are lost equal to the number on the unit wheel, but this is of small consequence in a fast running engine.

Each of these small horizontal shafts moves a pointer round a graduated dial, each division of which represents one revolution of the engine, the whole graduation being 100. As the pointers are superposed and in parallel plans, and difference in the speed of the engines is indicated by the faster engine's pointer running away from the other, there is a clutch in each shaft for the purpose of throwing the pointers out, so that the engines may be quickly regulated by stopping the faster pointer until the other catches up, and then noting the relative movement.

Ventilating fans.—In the engine room there is a fan manufactured by the Boston Blower Co., with ducts so arranged that air can be forced into the engine room or exhausted from it. The fan for ventilating the living spaces in the ship is of the same type, arranged to exhaust the air. The engines for these fans are all alike, double, and coupled directly to the fans.

Diameter of engine cylinders (2)	3½ inches.
Stroke of engine cylinders (2)	2½ inches.
Diameter of engine-room fan	24 inches.
Width of engine-room fan	9 inches.
Diameter of living-space fan	36 inches.
Width of living-space fan	10 inches.

Boilers.—The two boilers are of the horizontal, fire-tubular, straightway or gunboat type, each having two corrugated furnaces with a separate combustion chamber for each furnace. The front and back heads are curved at the top to reduce the weight of longitudinal bracing. Each boiler has a self-closing main and auxiliary stop valve, two feed check and stop valves, two glass water gauges, four gauge cocks, a bottom and surface blow valve, one safety valve, one sentinel valve and a self-closing stop valve for dynamo engines. The Craig circulating apparatus is fitted to each boiler. In this the feed water is delivered through a small nozzle, inducing a current of water from the bottom of the boiler around this nozzle. Branch pipes are fitted for steam connections, so that the water may be circulated when raising steam if there is steam in one boiler. Besides this, the auxiliary feed pump can be used to circulate, pumping from the bottom blow and delivering through auxiliary feed checks.

Steam pressure	160 pounds.
Length of boilers	17 feet 0 inch.
Diameter, outside	8 feet 9½ inches.
Grate surface, total, square feet	87.75
Heating surface, total, square feet	2,686.
Thickness of shell	¾ inch.
Thickness of butt straps	¾ and $\frac{9}{16}$ inch.
Thickness of heads	¾ inch.
Combustion chambers, thickness of plates	½ inch.
Combustion chambers, width at top	3 feet 6 inches.
Combustion chambers, depth	3 feet 0 inch.
Combustion chambers, width of water space between	6 inches.
Tube sheets, thickness	¾ inch.
Furnaces, greatest internal diameter	43 inches.
Furnaces, least internal diameter	39 inches.
Furnaces, length	7 feet 5½ inches.
Furnaces, length of grate	6 feet 9 inches.
Tubes, number in each boiler	314.
Tubes, diameter outside	2 inches.
Tubes, length between tube sheets	7 feet 0 inch.
Heating surface, tubes	1,173 square feet.
Heating surface, combustion chambers	85 square feet.
Heating surface, furnaces	85 square feet.
Heating surface, total for each boiler	1,343 square feet.
Ratio of heating to grate surface	30.61
Area through tubes, square feet	5 44
Ratio of grate surface to area through tubes	8.06
Diameter of safety valve (one)	3.5 inches.
Diameter of longitudinal braces	2¼ inches.
Number	7.
Diameter of smoke pipe	4 feet 7 inches.
Height above grate	40 feet.

Forced draft.—The forced draft is on the Kafer ash-pit system, air being supplied by two fans placed on the coal-bunker bulkheads adjacent to the boilers, one fan for each boiler. They take air from ducts over the tops of the boilers, and deliver through pipes beneath the fire-room floor to the ash-pits and furnace fronts. Suitable gear is fitted to the furnace doors so that they can not be opened until the draft is shut off. The engines for the fans are of the same design and same size as the engine-room fan.

Diameter of fan engine cylinder (two)	3½ inches.
Stroke	2½ inches.
Diameter of fan	36 inches.
Width of fan	12 inches.
Diameter of inlet	22 inches.
Air ducts to each fan	2 feet 6 inches × 6½ inches.

There is a small workshop, with lathe and vise bench, the lathe run by a Crist engine.

The ash-hoist is a simple steam cylinder, which takes steam at the top in hoisting and exhausts in lowering. The transmission of power is by wire rope. The cylinder is 7 inches in diameter, and has a stroke of about 52 inches.

Steering Apparatus.—The steering engine is one of Williamson's, and is placed beneath the water-tight deck, just forward of the tiller. It is operated in the usual manner from the pilot house on the bridge, and by gearing from the poop or from the hand-steering wheels under the poop. There are electric indicators and telegraphs from and to the engine, on the bridge, in pilot house, on the poop and at the hand-steering wheels under the poop. The engine is double, with cylinders 6 inches diameter by 6 inches stroke.

The capstan engine is of the well-known Providence type. It is a double engine, with cylinders 6 inches diameter by 8 inches stroke.

Electric Light.—The vessel is lighted throughout with electric light, and has, in addition, a search light on the bridge. The plant consists of two Thompson-Houston dynamos, each of 80 volts and 50 ampères, each operated by an Armington & Sims' vertical engine, designed for a working pressure of 80 pounds per square inch. The engines are double, with cranks opposite each other, and run at 500 revolutions per minute. The cylinders are each 5 inches diameter and 3 inches stroke.

Coal Bunkers.—There are seven bunkers below the water-tight deck, one forward of the boilers, two on each side of them, and one outboard of each main engine. There are, in addition, six bunkers above this deck which trim into the lower ones, and three compartments between these bunkers intended for the stow-

age of patent fuel. Much of the latter space has necessarily been taken up by pipes communicating with the forward part of the ship, and will generally not be of use for the stowage of coal, except for a small quantity in bags. The capacity of the bunkers proper is about 115 tons, but they are difficult to stow. The same system of coaling obtains as in vessels of the *Baltimore* class, that is through chutes in the side of the ship, and trunks inside with water-tight doors in the protective deck, and slide doors to the bunkers above this deck; the trunks are on a small scale, more in proportion to the size of the ship than to that of the men, and, as each bunker is small, coaling is a very slow process.

WEIGHTS.

Propelling machinery, including all the auxiliary machinery hereinbefore described (except capstan, steering, ship ventilating, workshop, and dynamo engines), together with water in boilers, pipes, and condenser ..... 124.46 tons.

The most important detailed weights are:

	Pounds.
Foundations	1,773
Bed-plates, shaft bearings, brasses and caps, stern tubes, and outboard bearings	8,884
Frames and cross-head guides	3,161
Cylinders, bushings, covers, stuffing boxes, relief valves, drain cocks, etc	11,906
Pistons, piston-rods and nuts	1,272
Cross-heads	1,150
Connecting rods	1,512
Valves and stems	1,109
Valve and reversing gear and engines	3,699
Crank-shafts and bolts	2,895
Thrust, stern and propeller shafts and casing	12,034
Propellers	4,604
Throttles	504
Main steam pipes, bleeders and pass overs	829
Exhaust pipes and valves	1,268
Main condenser, valves and nozzles	8,821
Circulating pump and connections	1,868
Air pump and connections	2,723
Injection and discharge valves	2,664
Feed tank, vapor pipe, etc	1,206
Main feed pump, pipes and valves (including checks)	2,023
	75,905

Boilers, bars, furnace doors, blow valves, Kingston, dry pipes	94,681
Water in boilers (6 inches above tubes)	44,840
Main and auxiliary stop valves	1,368
Water gauges, cocks and pipes	539
Safety valves and escape pipes	1,192
Blowers, air ducts, etc	5,204
Smoke-pipe, uptakes and doors	16,803
	164,627
Boiler and pipe covering	4,700
Boiler circulating apparatus	531
Auxiliary feed pump and connections	2,449
Water service pump and connections	1,257
Fire and bilge pump and connections	2,496
Engine room fan and ducts	1,422
Turning engine and gear	794
Distilling plant	3,673
Auxiliary condenser and connections	3,007
Auxiliary injection and discharge valve	674
Engine room floor and ladder	2,847
Fire room floor and ladder	3,109
Ash-hoisting engine, gear and railway	800
	27,759

TRIAL.

More than ordinary interest attached to this trial from the fact that it was the first one to be made according to the plan proposed by Engineer-in-Chief Melville (see JOURNAL, vol. II, p. 73), and the first in which the Weaver Revolution Recorder, described on p. 86 of vol. III of the JOURNAL, was used. Briefly, the method consists in running the vessel over a measured-mile course at a number of different speeds, and from the revolutions and speed determined on the mile a curve of revolutions per minute and speed is drawn for the vessel. The time or official run is then made, and from the mean revolutions per minute for this run the average speed for the trial is read off from the curve. The instructions to the Trial Board were to make one double run at seven knots, one at nine, two at eleven, two at twelve and two at a higher speed, and to construct the curve from the spots so determined. It may be well to note here that the Weaver recorded each second of time on the mile, the beginning and the

end of each run, and the total number of revolutions of each screw. An observer was stationed on the forecastle and another on the poop, each of whom, by means of a push button, recorded on the tape of the "Weaver" the instant of passing the ranges at each end of the course. The observations were thus as accurate as it is possible to make them, the only chance for variation being in the position occupied by the observer in sighting the ranges. This is manifestly very small, but in order to reduce it as much as possible I would suggest that fixed revolving-chair seats be provided, which would insure the observer always being in the same spot. I think this would also contribute to greater accuracy in making the electric contact for the Weaver, for the contact piece could be held in the hand and not fastened to the rail, as was the case in these trials.

Owing to a desire to take advantage of the good weather, the mile trials were made on two days, the 21st and 23d of January, and on each of the days there was a change of tide during runs, as shown in Nos. 3 and 4 and 8 and 9 of table I.

On the four-hour run the revolutions of the engine were, according to the Board's instructions, taken from the engine-room counters, of which two were fitted for each engine. One of those for the starboard engine carried away early in the trial, and the revolutions for that engine were taken from one counter only; but as it was a rotary counter there is no doubt that it did not slip. Observations were taken from each counter every 15 minutes, the results of which are given in table II.

Table I and diagram give the data for the progressive trials on the measured mile.

From these data the curve of speed and revolutions was plotted on a large scale, which would enable hundredths of a knot to be measured accurately. This curve, on a reduced scale, is annexed.

For convenience during the preliminary trials a diagram was prepared by Mr. A. H. Raynal, superintendent of the works, by means of which the I.H.P. of each cylinder could be read off as soon as the mean effective pressure was known for each cylinder. The constant of each cylinder being known, and multiplied by

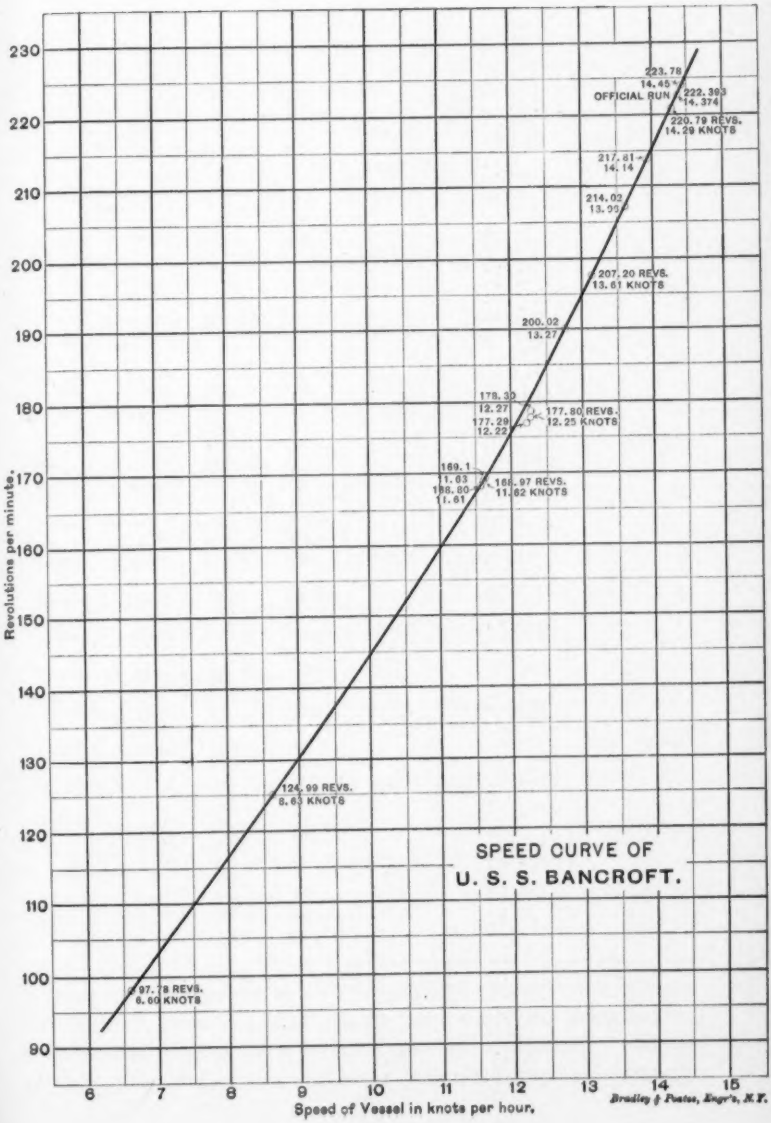

Revolutions per minute.
230
220
210
200
190
180
170
160
150
140
130
120
110
100
90
6
7
8
9
10
11
12
13
14
15
Speed of Vessel in knots per hour.
223.76
14.45
OFFICIAL RUN
222.393
14.374
220.79 REVS.
14.29 KNOTS
217.81
14.14
214.02
13.92
207.20 REVS.
13.61 KNOTS
200.02
13.27
178.30
12.27
177.29
12.22
177.80 REVS.
12.25 KNOTS
169.1
11.63
166.80
11.61
166.97 REVS.
11.62 KNOTS
124.99 REVS.
8.63 KNOTS
97.78 REVS.
6.60 KNOTS
SPEED CURVE OF
U. S. S. BANCROFT.
Bradley & Peatse, Engr's, N.Y.

Table I.

RECORD ON MEASURED MILE BY WEAVER APPARATUS.

No. of run.	Time on the mile.	Revolutions on the mile.			Revs. per min. (mean both shafts.)	Mean speed knots.	Second Means.	
		Starboard.	Port.	Mean of both shafts.			Speed knots.	Revs. per min.
	min. sec.							
Up.	10 13·69	1,006.24	1,010.62					
1 Down.	7 56·90	767.86	769.94					
Mean.	9 05·30	887.06	890.29	888.67	97.78	6.60	6.60	97.78
Up.	7 12·23	905.32	885.90					
2 Down.	6 41·59	852.79	829.87					
Mean.	6 56·91	879.06	857.89	868.47	124.99	8.63	8.63	124.99
Up.	5 14·17	873.84	877.56					
3 Down.	5 05·88	857.26	879.96					
Mean.	5 10·02	865.55	878.76	872.16	168.80	11.61		
Up.	5 10·92	863.25	871.63				11.62	168.97
4 Down.	5 07·88	872.15	882.23					
Mean.	5 09·40	867.70	876.93	872.32	169.15	11.63		
Down.	5 02·95	900.08	886.99					
5 Up.	4 46·36	847.54	848.25					
Mean.	4 54·66	873.81	867.62	870.72	177.29	12.22		
Down.	5 03·91	898.92	897.85				12.25	177.80
6 Up.	4 42·85	845.81	844.92					
Mean.	4 53·38	872.37	871.39	871.88	178.30	12.27		
Down.	4 35·03	927.53	928.40					
7 Up.	4 27·68	882.92	879.46					
Mean.	4 31·35	905.22	903.93	904.57	200.02	13.27		
Down.	4 20·99	932.64	929.93				13.61	207.02
8 Up.	4 14·84	910.73	906.62					
Mean.	4 17·91	921.69	918.28	919.98	214.02	13.96		
Down.	4 12·33	916.22	928.12					
9 Up.	4 16·90	926.54	926.25					
Mean.	4 14.61	921.38	927.19	924.28	217.81	14.14		
Down.	4 05·76	912.90	912.64				14.29	220.79
10 Up.	4 12·66	946.47	945.41					
Mean.	4 09·21	929.69	929.02	929.36	223.78	14.45		

a mean effective pressure and revolutions greater than would be obtained, gives an I.H.P. point from which to draw a line through the point of zero pressure and no revolutions. Dividing the horizontal and vertical scales proportionally and drawing

lines to the origin, the points where these lines intersect the revolution ordinates give the I.H.P. corresponding to the revolutions and pressure. To be of practical utility the diagram should be laid down on a large scale, and I think that a second one ranging from zero to 10 pounds mean effective pressure would facilitate the work. Diagrams of this kind prepared on board ship would help greatly the quarterly computations of I.H.P., unless a table of such values has already been prepared, and would have the advantage of eliminating errors, provided only that the one upper point is correct.

The accompanying diagram is for the H.P. cylinder of the *Bancroft.*

FOUR-HOUR TRIAL.

The official contract trial was made on January 26, 1893, and consisted of a four-hours' straightaway run almost south from Beaver Tail light. The mean number of revolutions, as shown by table II, was 222.393 corresponding to a mean speed of 14.374 knots:

During the contract trial two patent taffrail logs were used, each with 400 feet of line. Both were new, but the one referred to as *B*, in the following table, had been used on the trip of the *Bancroft*, from the contractor's works at Elizabethport, N. J., to Newport. It had underlogged the vessel as shown by comparison with bearings. These logs were not tested on the measured mile, but on the way out of Narragansett Bay, before the contract trial, their readings were taken for a distance known by bearings to be about two knots. On this distance log A was correct, while log B was 0.2 knot short.

The readings for the four-hour trial were as follows:

	Log A.	Log B.
1st hour, ...	14.7	13.5
2d "	14.7	13.6
3d "	14.6	13.5
4th "	14.7	13.5
Average, ...	14.675	13.525

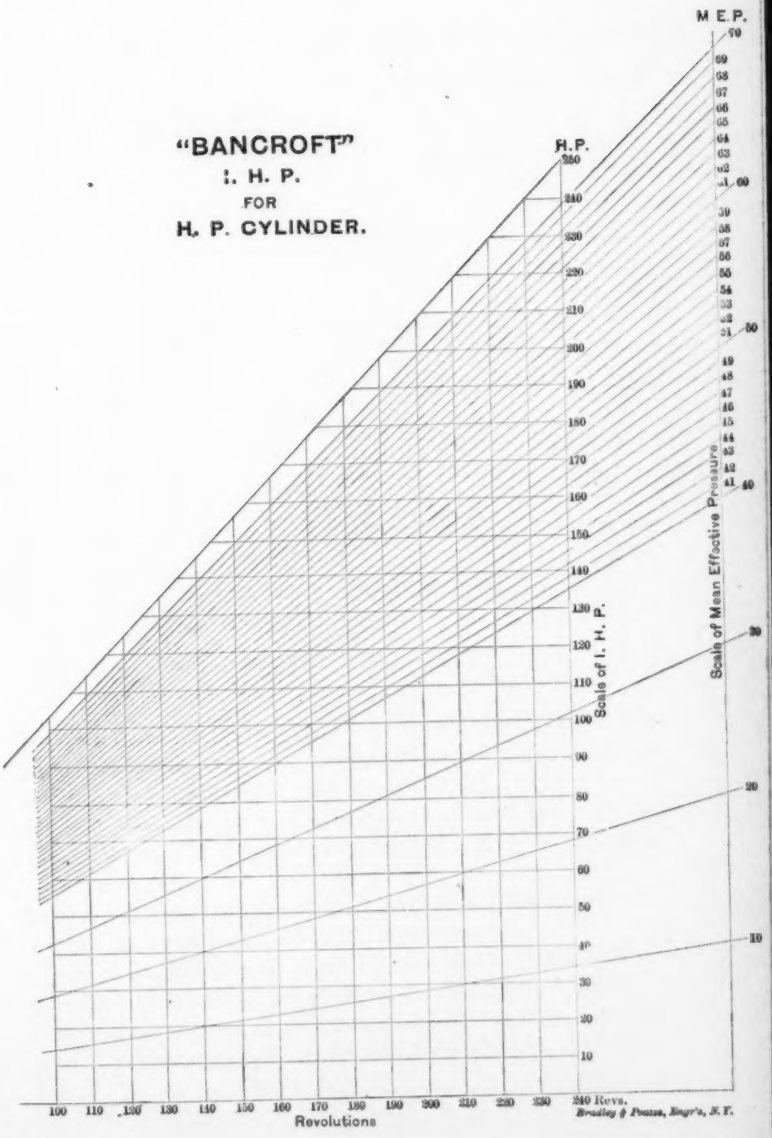

"BANCROFT"
I. H. P.
FOR
H. P. CYLINDER.
M E.P.
H.P.
Scale of Mean Effective Pressure
Scale of I. H. P.
Revolutions
100 110 120 130 140 150 160 170 180 190 200 210 220 230 240 Revs.
Bradley & Pease, Engr's, N.Y.

Table II.

REVOLUTIONS FOR FIFTEEN-MINUTE PERIODS OF FOUR-HOUR OFFICIAL TRIAL.

	Average revolutions per minute for the fifteen minutes ending at this time—			Remarks.
	Starboard.	Port.	Mean.	
A. M. 8·45	228.066	225.00	226.53	
9·00	226.60	224.80	225.70	Counter readings at 8.45 A. M.:
9·15	224.33	226.33	225.33	Starboard, 224809;
9·30	223.27	224.07	223.67	Port, . . 224156.
9·45	220.13	222.27	221.20	Counter readings at 12.45 P. M.:
10·00	224.20	223.00	223.60	Starboard, 278352;
10·15	222.47	221.93	222.20	Port, . . 277362.
10·30	222.06	222.10	222.09	Total revolutions in four hours:
10·45	221.93	221.33	221.63	Starboard, 53543
11·00	219.60	219.67	219.63	Port, . . . 53206
11·15	224.20	221.07	222.63	
11·30	225.27	219.80	222.53	Mean, . 53374·5
11·45	225.30	220.80	223.05	Average revolutions per minute for the trial:
P. M. 12·00	225.07	221.27	223.17	$\frac{53374\cdot5}{240} = 222.393$
12·15	221.27	218.67	219.97	
12·30	221.70	219.67	220.68	
12·45	222.14	220.27	221.20	
Means.	223.09	221.69	222.392	

The trial was a marked success in every way. The steam pressure and revolutions were maintained with marked uniformity, and the machinery gave entire satisfaction, not a drop of water being used on any of the bearings. There was scarcely any vibration, a glass of water, evenly full, standing on the cabin deck without spilling.

The water at the bow rose about a foot with a flat top and no spray. At the stern there was very little disturbance.

The speed called for in the contract was 12 knots, the boilers being worked under an air pressure of one-half inch of water.

In explanation of the remarkable success shown on the trial above the contract requirements, it may be said that, when the

Bancroft was designed, it was expected that the air pressure would be about an inch, as in the case of the *Yorktown*, and all the details of the design were based on this. After the design had been made, the air pressure was reduced, and it became necessary to fix upon a new contract speed. As no data were available of accurate trials with half an inch air pressure, an estimate was based on the *Yorktown's* performance with an inch. It was supposed that about twelve knots and a half could be obtained, if everything worked perfectly, and, in accordance with the Department's practice of allowing a slight margin for unforeseen contingencies, twelve knots was fixed as the contract speed.

The *Bancroft* has a remarkably fine hull, the lines being the design, it is understood, of the late Naval Constructor Gatewood. This accounts for a part of the superiority over the *Yorktown's* record. In addition, the contractors used special machinery, with which they have been building high-speed stationary engines, for making all working parts absolutely true, the piston and valve rods and shafting being all "ground true." Selected Pocahontas coal was also used, while the *Yorktown* had anthracite. Finally, the *Bancroft's* engines are vertical, while those of the *Yorktown* are horizontal.

DATA OF TRIAL.

Draught, moulded, at beginning,	Forward	11 feet 0 inches.	11 feet 6 inches.
	Aft	12 feet 0 inches.	
Draught, moulded, at end,	Forward	10 feet 10 inches.	11 feet 4½ inches.
	Aft	11 feet 11 inches.	
Draught, mean, for trial,	Forward	10 feet 11 inches.	11 feet 5¼ inches.
	Aft	11 feet 11½ inches.	

Displacement at mean draught on trial	832 tons.
Area of immersed midship section	277 square feet.
Wetted surface	6,150 square feet.
I.H.P. (total) per 100 square feet wetted surface	19.24
I.H.P. (total) per 100 square feet wetted surface at 10 knots, reduced in ratio of 3.5 power	5.40
Average speed, in knots	14.374
Slip (mean of both screws), per cent	15.50
Speed ^{3} × area immersed midship section ÷ I.H.P.	711
Speed ^{3} × displacement $^{\frac{2}{3}}$ ÷ I.H.P.	219

SYNOPSIS OF STEAM LOG.

	Starboard.	*Port.*
Revolutions of main engines per minute	223 09	221.77
Piston speed, in feet per minute	743.5	739.2
Steam pressure at boilers (per gauge)	167.56	
Steam pressure at engines (per gauge)	163.00	163.06
Steam pressure at first receiver (absolute)	64.32	65.76
Steam pressure at second receiver (absolute)	24.10	25.76
Vacuum in condenser (inches of mercury)	26.34	
Opening of throttle	Wide.	Wide.
Steam cut off in fraction of stroke from beginning, H.P.	.72	.72
Steam cut off in fraction of stroke from beginning, I.P.	.72	.72
Steam cut off in fraction of stroke from beginning, L.P.	.72	.72
Double strokes of air pump per minute	75.5	
Double strokes of feed pump per minute	35.25	
Revolutions of circulating pump per minute	236.31	
Temperatures in degrees Fahrenheit, Engine room	76.2	
Temperatures in degrees Fahrenheit, Injection	44.0	
Temperatures in degrees Fahrenheit, Discharge	87.7	
Temperatures in degrees Fahrenheit, Hot well	90.6	
Temperatures in degrees Fahrenheit, Feed	119.9	
Temperatures in degrees Fahrenheit, Fire room	77.5	
Revolution of blowers per minute	574.	580.
Mean pressures in cylinders, H.P	61.98	65.87
Mean pressures in cylinders, I.P	21.77	22.93
Mean pressures in cylinders, L.P	13.98	12.48
Mean pressures in cylinders, Aggregate equivalent reduced to L.P	34.61	35.28
I.H.P. H.P. cylinder	197.00	208.04
I.H.P. I.P. cylinder	169.09	176.52
I.H.P. L.P. cylinder	222.33	210.36
I.H.P. Collective, each main engine	588.42	594.92
I.H.P. Collective, both main engines	1,183.34	
I.H.P. Air-pump	5.09	
I.H.P. Circulating pump	1.35	
I.H.P. Feed pump	1.70	
I.H.P. Blowers (estimated)	12.00	
I.H.P. Other auxiliaries (estimated)	9.50	
Collective I.H.P. of all auxiliaries	29.64	
Collective I.H.P. of all machinery in operation	1,212.98	
Indicated thrust (main engines only), pounds	11,232.	11,422.
Indicated thrust, per square foot of developed area (15.00 square feet) of propeller, pounds	748.7	761.4
Indicated thrust per square inch of thrust bearing, pounds	42.50	43.22
Cubic feet swept per minute by L.P. piston per I.H.P	6.60	6.49
Square feet of cooling surface per I.H.P	1.54	

	Starboard.	*Port.*
Square feet of heating surface per I.H.P.	2.18	
I.H.P. per square foot of G S (all machinery in operation)	13.82	
I.H.P. per ton of boilers, water, fittings, and all fire-room weights (76.98 tons)	15.76	
I.H.P. per ton of propelling machinery, boilers and water (124.46 tons)	9.75	

From the observed speeds and revolutions on the progressive trials and the I.H.P. computed from the indicator diagrams taken at the same time, a series of spots was obtained through which fair curves have been drawn, as shown in the accompanying diagram (Fig. 2).

From these curves the following table is constructed:

Table III.

Revolutions per minute.	Speed in knots.	I.H.P. Starboard.	I.H.P. Port.	I.H.P. Total.
75	4.85	30.00	35.00	65.00
100	6.80	50 00	60.00	110 00
125	8.60	90.00	110.00	200.00
150	10 40	158.00	182.00	340.00
175	11.95	255.00	285.00	540.00
200	13.30	385.00	430.00	815.00
225	14 52	565.00	605.00	1,170.00

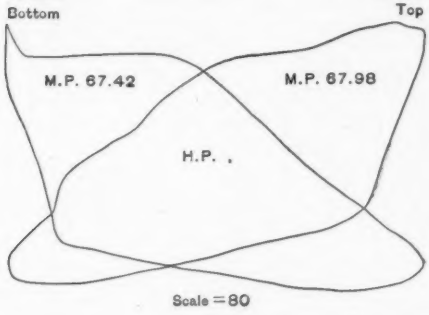
Figs. 3 and 4 are the graphic logs of the two engine rooms during the trial. It is to be noted that the various scales used were selected merely for convenience in laying down the values, except that the mean pressure scales are approximately proportional to the areas of the cylinders, that is, as the L.P. is about five times the area of the H.P., one pound in the L.P. has an ordinate five times that for one pound in the H.P.

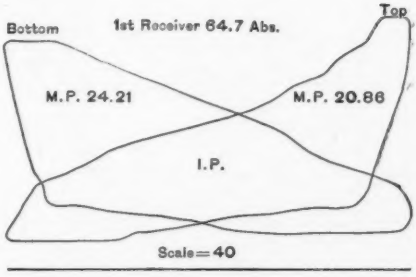
Fig. 5 shows a set of cards from the starboard engine.

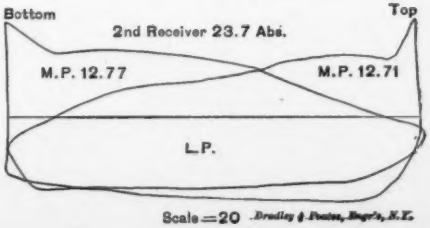
U. S. S. BANCROFT.

Starboard Engine.

January 26th, 1893.	11.45 A.M.
Steam Pressure at Eng	164 lbs.
Revolutions Per Min.	225.07
Vacuum	26 Ins.







Bradley & Peaine, Engr's, N.Y.

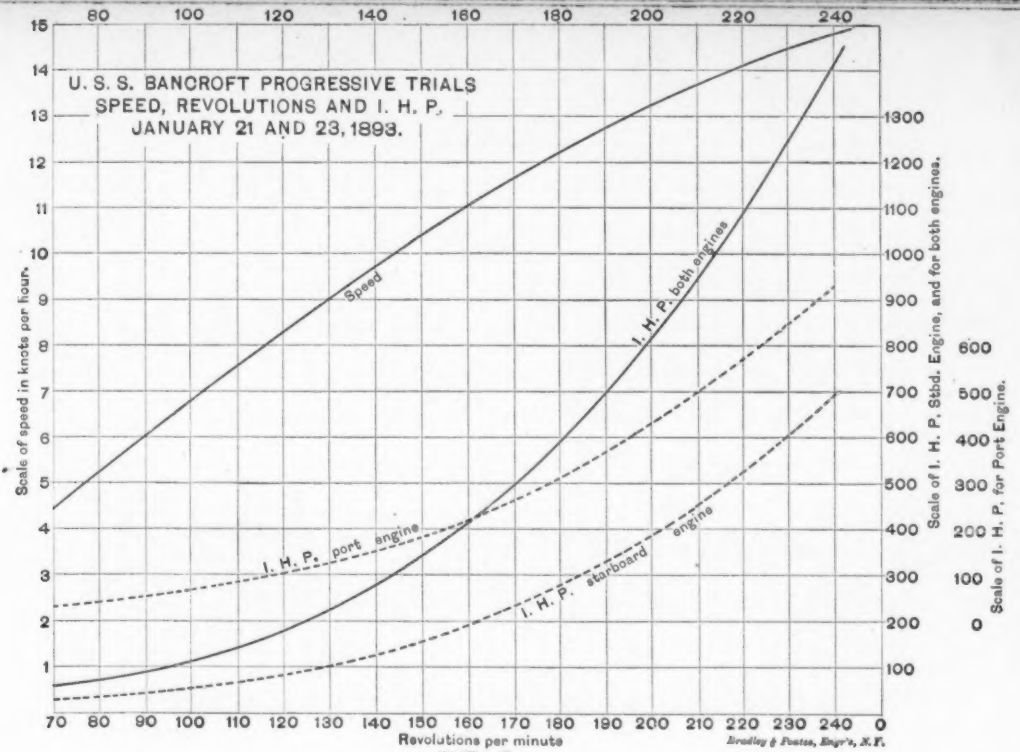

U. S. S. BANCROFT PROGRESSIVE TRIALS
SPEED, REVOLUTIONS AND I. H. P.
JANUARY 21 AND 23, 1893.
Speed
I. H. P. both engines
I. H. P. port engine
I. H. P. starboard engine
Scale of speed in knots per hour.
Scale of I. H. P. Stbd. Engine, and for both engines.
Scale of I. H. P. for Port Engine.
Revolutions per minute
Bradley & Peatse, Engr's, N.Y.

FIG. 2

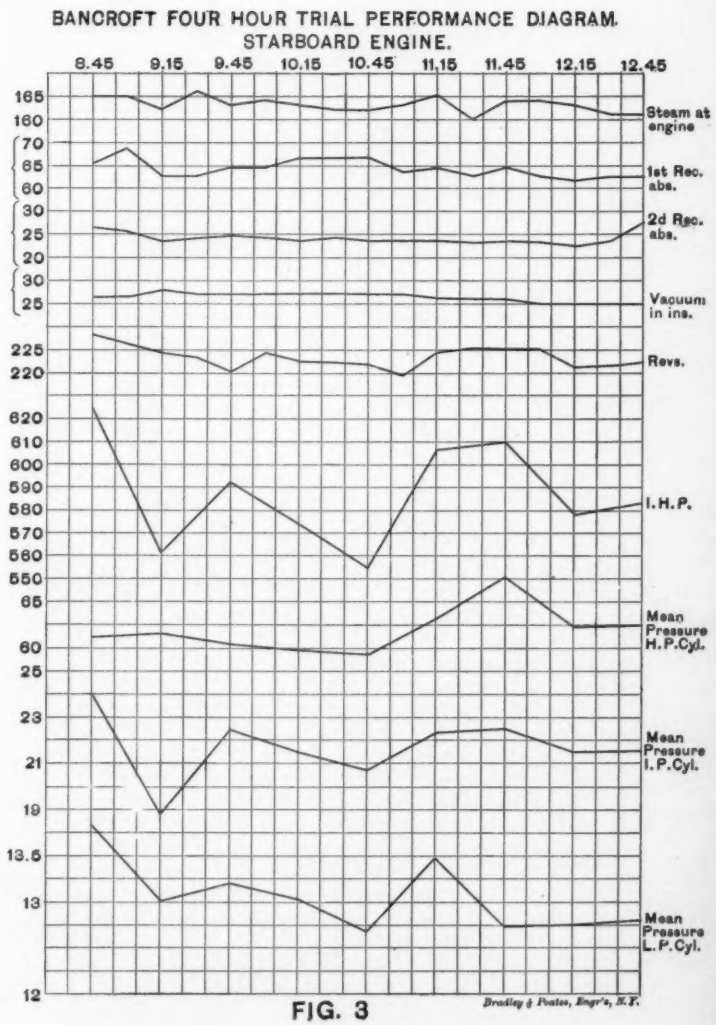

BANCROFT FOUR HOUR TRIAL PERFORMANCE DIAGRAM.
STARBOARD ENGINE.
8.45
9.15
9.45
10.15
10.45
11.15
11.45
12.15
12.45
165
160
70
65
60
30
25
20
30
25
225
220
620
610
600
590
580
570
560
550
65
60
25
23
21
19
13.5
13
12
Steam at engine
1st Rec. abs.
2d Rec. abs.
Vacuum in ins.
Revs.
I.H.P.
Mean Pressure H.P.Cyl.
Mean Pressure I.P.Cyl.
Mean Pressure L.P.Cyl.
Bradley & Pentes, Engr's, N.Y.

FIG. 3

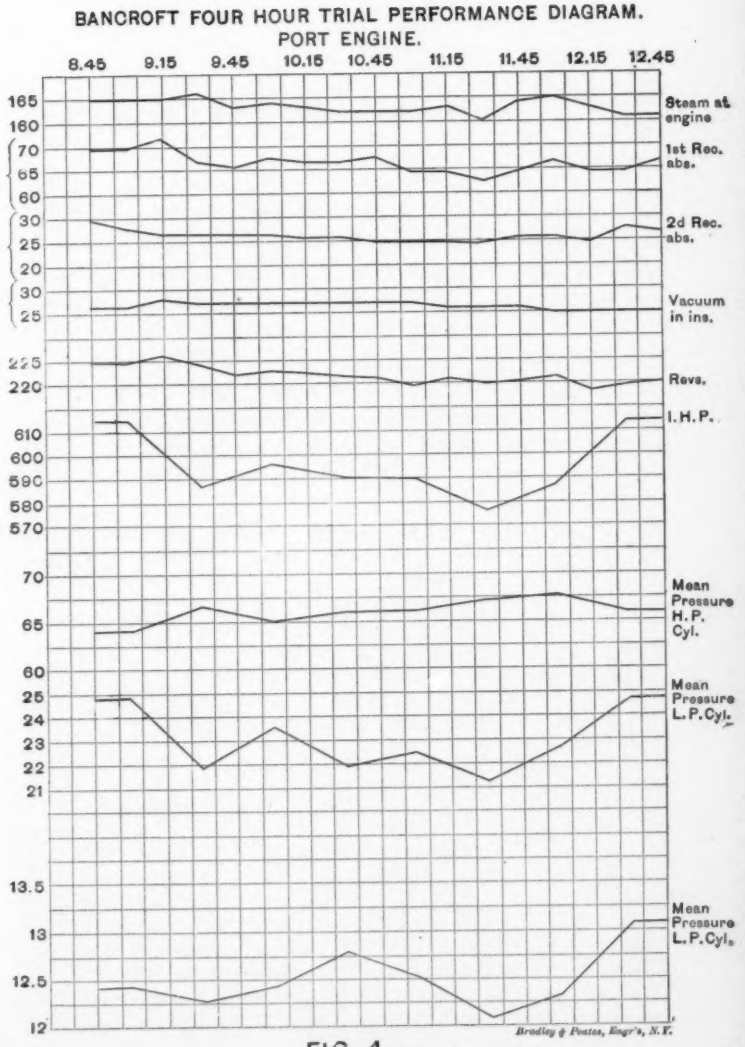

BANCROFT FOUR HOUR TRIAL PERFORMANCE DIAGRAM.
PORT ENGINE.
8.45
9.15
9.45
10.15
10.45
11.15
11.45
12.15
12.45
165
160
70
65
60
30
25
20
30
25
225
220
610
600
590
580
570
70
65
60
25
24
23
22
21
13.5
13
12.5
12
Steam at engine
1st Rec. abs.
2d Rec. abs.
Vacuum in ins.
Revs.
I.H.P.
Mean Pressure H.P. Cyl.
Mean Pressure L.P.Cyl.
Mean Pressure L.P.Cyl.
Bradley & Pontas, Engr's, N.Y.

FIG. 4

THE CUNARD STEAMER *CAMPANIA*.

The placing of this splendid vessel in regular service has been awaited with great interest by all who pay any attention to shipping. Every notice which pretended to give an accurate description has been eagerly read, but until recently all the accounts omitted any definite data of the machinery.

The number of *Engineering* for April 21st, which is reviewed elsewhere in this issue, makes up for the shortcomings of all previous notices, and the following account consists of extracts from it. The detailed description may be preceded by a summary of the principal dimensions:

Length over all	622 feet.
Length between perpendiculars	600 feet.
Breadth, extreme	65 feet 3 inches.
Depth from upper deck	41 feet 6 inches.
Depth from shade deck	59 feet 6 inches.
Draught, mean	25 feet 0 inches.
Tonnage	12,950 tons.
Displacement (about)	18,000 tons.
Indicated horse power (about)	31,000.
Speed	23.18 knots.
Revolutions of engines	84.
Passenger accommodation { 1st class	600.
2d class	400.
3d class	700 to 1,000.
Crew	415.
Cargo carrying capacity	1,620 tons.
Capacity coal bunkers	3,500 tons.
Twin screw engines:	
High-pressure cylinders, (4) diameter	37 inches.
Intermediate cylinders, (2) diameter	79 inches.
Low-pressure cylinders, (4) diameter	98 inches.
Stroke	69 inches.

Boilers:

Twelve double-ended	diameter 18 feet. length... 17 feet.
One single-ended	diameter 18 feet. length... 11 feet.
One single-ended	diameter 10 feet. length... 10 feet.
Total number of furnaces	102.
Working steam pressure	165 pounds.

The enormous size of the vessel brought up a number of problems, among which was that of the lengths of hull plating. It was finally decided that the minimum length of plates used in the shells of the ships should be 25 feet and the breadth 6 feet. Each plate thus weighed over 2 tons. The bulk of the plates used are of this size, but in many cases, notably below the machinery, the width was increased to nearly 8 feet. The thickness varied from $\frac{3}{4}$ inch to 1 inch. The frames used were of the channel-bar section, and the beams of the ordinary Butterley bulb section.

The first keel plate of the *Campania* was laid on September 22, 1891. The keel is not of the bar external form, which is now being discarded in most large ships, but is entirely internal. It is built of plates 4 feet 6 inches deep, extending right fore-and-aft, and having on the upper and lower edges at either side very strong angles. The keel plate was formed of a broad plate 54 inches by 1 inch, with a doubling plate about $\frac{7}{8}$-inch thick on the inside. These plates are connected to the bottom of the vertical internal-keel plate, and together they form the main center-line girder of the ship.

Under the engines this girder is increased not only in strength but in depth to about 8 feet, and forms the base for the engine seating. The keel was set on a grade of $\frac{7}{16}$ inch to 1 foot, and before it was completely laid the work of constructing the double bottom was proceeded with, the first double-bottom frame being erected about three weeks after the first keel plate was placed in position. Instead of carrying the ordinary frames or ribs of the ship from the keel to the top decks, as is usual in small craft, the bottom or floor of the ship is supported on heavy longitudinal

girders extending from end to end of the ship, and between these again runs a series of intercostals. In addition to the center-line girder already referred to, there are four built-up I-girders, two about 15 feet on either side of the center line at the middle of the ship and two close to the bilge, or about 30 feet on either side of the keel, these latter forming the outer edges of the double bottom. These girders are of vertical plates with double angles like the center one. In the intermediate spaces between the longitudinal girders are two pairs of intercostals fitted between the floor plates and carrying these up to their work.

As in the case of the center-line girder, each of the other girders is of greater depth under the machinery space, extra longitudinals being also fitted in this part to give the necessary strength for the carrying of the engines. The whole of the riveting within the range of this double-bottom framing was done by hydraulic machinery. The extreme depth of the girders under the machinery space involved difficulties in the carrying out of this idea, but as much importance was attached to the absolute rigidity of the construction under the engines, special appliances, involving new large riveting machines, were obtained. To insure perfect workmanship, too, the holes in all cases under the machinery space were drilled or rimered perfectly true, while the angle irons were joggled and welded, forming continuous frames. The frames extend from a margin plate on the outer girders to the upper deck. The frames are of channel section, and thus combine the ordinary angle and reverse frame in one section, without the necessity of joining the two separate angles together.

For the purpose of working these channel bars, the Fairfield Company had a special tool constructed which we illustrated recently (see "Engineering," vol. lv., page 155). The frames are placed 30 inches apart, and extend in one length from the margin plate of the double bottom to the upper deck stringer plate, and to the poop and forecastle stringers at the ends of the ship. The maximum length of channel bar used was about 50 feet, in one continuous length. These channel frames were strengthened at intervals fore and aft the ship by solid web frames,

or partial bulkheads. These web frames are built up of plates with double angles on the inside edge, and connected by double angles to each of the deck stringers. They vary from 2 feet to 3 feet in breadth. In the machinery spaces, again, these web frames are greatly increased in number, the channel frames being also of heavier section in the wake of the engines.

Upon each alternate channel frame were fastened the Butterley bulb beams carrying the deck platforms. These beams are of the largest section that has yet been rolled for this purpose. The orlop, lower, main, and upper decks were constructed with the main scantlings, whilst above these the scantlings tapered on the promenade, forecastle, and poop decks, and the shade or boat deck. In the case of the orlop deck beams it was considered necessary, on account of the space for the boilers and machinery, to make the stringers of special form to maintain the stiffness of the structure. The plan adopted was to use an open girder instead of the old-fashioned box girder, which, being permanently closed, prevents inspection or repainting. The open girders are built up with double stringers having brackets above and below, with diaphragm plates between, and strong face angles on the inner edge. On the lower, main and upper decks the stringer plates take the ordinary form, with the exception that the connection to the shell plate is made by double angles instead of single angles as is usual.

The plating of the ship is arranged on the overlap system from the keel strake up to the underside of the main deck sheer strake, a system of plating first inaugurated, we believe, in smaller cargo vessels on the northeast coast, but since adopted in all types of steamers as being a most suitable butt connection. The butts of the upper and main sheer strake and the strake between are connected by double butt straps—inside and out. In connection with this we may note the somewhat new feature indicated by the round-headed rivets along the upper edge of the sheer strake as evidence of a marked advance in the structural connections of large ships—namely, that the whole connection between the upper deck with its stringer plate and heavy deck plating is fixed to the sheer strake by strong angles riveted together, in this case

by hydraulic machinery, instead of by hand, which would have given a much less efficient job, owing to the heavy plates in use. And although at first sight these projecting heads may detract from the graceful lines of the ship, the advantages are more than compensatory. Again, there are no doubling plates on the bilges of the ship, where leakage often takes place, with the resultant deterioration due to the water getting between the two thicknesses. The doubling plate has been dispensed with by increasing the thickness of the plates themselves, and by adopting a treble, instead of a double-riveted landing, which greatly increases the protection against any possible "working" at this point.

The important matter of bulkhead subdivision had very full consideration in the working out of the design of the ships. About this time the report of the Bulkhead Committee* had been published, and it was deemed advisable to fulfil all the suggestions made by the committee as far as they were practicable. The subdivision is by thwartship bulkheads only, and of these there are eighteen. Center-line bulkheads have not been adopted, as they would have interfered seriously with the arrangement of the boilders, and would have given no compensating advantage, for in the case of a ship divided by a center-line bulkhead, the flooding of a large compartment, say on the port side, as the result of an accident, would give the ship a heavy list, which is itself a danger, and would probably necessitate the filling of a corresponding compartment on the starboard side, so that the immersion of the ship would not be less than with a filled compartment the full width of the ship. The engine room, it is true, is divided by a center-line water-tight bulkhead having large doors on the lower platforms for the efficient control of the engines. The Fairfield Company have rather shortened the length of the compartments between the thwartship bulkheads. The maximum length is about 65 feet amidships—or only about 10 per cent. of the total length of the ship—while at the ends, more especially forward, where the risk from collision is greatest, the distance between bulkheads

* See "Engineering," vol. lii, page 392.

is considerably reduced. The subdivision of the vessel, in fact, is carried out to such an extent that any two, and in many cases even three, compartments might be flooded with water and the steamer remain in perfectly safe condition. Again, all the "vitals" of the ship—the machinery, boilers, auxiliary appliances, etc.—are in duplicate, so that the flooding of one or two compartments cannot well completely disable the propelling, lighting or navigating, or, indeed, any of the important machinery in the vessel. The construction of the bulkheads also follows on the suggestions of the Bulkhead Committee, they being strengthened by deep bars of channel section instead of by angle irons. From tests that have been made subsequent to the construction of the bulkheads, this method of stiffening has been found distinctly satisfactory. Where necessary, doorways have been made under the lower deck; but the number of these has been minimized, and they are in all cases fitted with water-tight doors. In the more vulnerable parts of the ship, however, these are dispensed with altogether below the main deck. As the vessels are classed as armed cruisers for service when required by the British Admiralty, they have been built to comply with their conditions as to strength, buoyancy under disastrous conditions, and coal endurance. They have, therefore, water-tight coal bunkers at the side of and over the top of the boiler compartments, forming a protection against the modern quick-firing gun.

About this time—the end of June—one of the most important parts of the structure was in the process of erection—the stern framing. The frame is of cast steel, and made by the Steel Company of Scotland, who deserve the greatest credit for the high quality of the material. In arranging the propeller brackets it was decided, with the view of bringing the propellers as near as possible to the center line of the ship, to have a small aperture. It has been suggested that the intention was, if found desirable, to use overlapping propellers; but we have authority for stating that no such intention was ever entertained by those responsible for the design of the *Campania* and *Lucania*.

The propellers are placed in the same line athwartship. The

sternpost is made up of four parts. The propeller brackets are machine-jointed on each side of the forward sternpost and connected thereto. The stern tubes are contained within the bossing, leaving them accessible for inspection at all times, whether the ship is at sea or in harbor. This is done by curving the shell plates and frames to suit the projection of the stern tube. The arrangement of the rudder is somewhat novel, although embodying no essentially new departure. In determining that the steering gear should be entirely under water, the better to conform to Admiralty requirements, it was felt that some concession should be made to ordinary appearance, without unnecessarily filling out the ship at the after end and interfering with the delivery of water from the propellers. A special arrangement was therefore devised by forming something like an annular chamber at the extreme after end, and having sufficient space within to receive the crosshead of the rudder, with its connecting rods leading into the body of the ship, and connecting thereto the steam steering gear, which we shall describe later when we come to deal with navigating appliances.

The rudder itself is of the ordinary center-fin plate pattern, with specially designed arms on either side, but more decidedly strengthened by webs on the top and bottom members, so that the rudder can be thrown hard over when the vessel is going at full speed. As the rudder had to be entirely below water, it became necessary to have a plate of exceptional size, not, perhaps, so much as regards length as to width. On trying nearly all the principal makers in this country it was stated to be impracticable to get such a wide plate; but subsequently the Fairfield Company got it from Messrs. Krupp, of Essen, the plate delivered measuring 22 feet by 11 feet 6 inches by 1¼ inches thick. The necessity for going to a foreign firm was afterwards called in question by several English makers, including, we understand, some of those who had actually been asked to supply the plate, and had returned the specification stating that it could not be done. It was matter of some surprise that the German maker could supply a plate the rolling of which British millowners, when asked, did not then care to undertake.

So soon as the contract was fixed, the work of designing the engines was started. The general arrangement of the engines is that first introduced in the North German Lloyd's steamer *Lahn*, built at Fairfield in 1887, and which has given so much satisfaction that the North German Lloyd's have had it adopted in later vessels built at Stettin. The Cunarders, however, are the first twin-screw steamers in which this type has been fitted. In this arrangement there are five cylinders, two high-pressure, one intermediate, and two low-pressure, but the great advantage of well-balanced parts of the three-crank engine is maintained by placing the high-pressure above the low-pressure cylinders, one pair of high-pressure and low-pressure being at the forward and the other pair at the after end of the engine. The adoption of five cylinders, too, reduces the diameter of each low-pressure cylinder to more moderate dimensions. The diameters of the cylinders are as follows: High-pressure (two), 37 inches; intermediate (one), 79 inches; and low-pressure, (two), 98 inches. The stroke of piston in each case is 69 inches.

From the base of the engines to the top of the cylinders the height is 47 feet. The Fairfield Company have adhered to the type of bed plate, column and condenser which has been adopted and proved satisfactory in so many of the notable vessels that have been constructed in their establishment. The bed plate is bolted down to the specially strong seating, which we have described in dealing with the construction of the hulls. The bed plate is 5 feet 6 inches deep, and is of cast iron. The castings are of the box type. The fixing of the engines to the ship at the bottom and the staying at the top has had special attention. The cylinders are braced together by heavy stays running fore and aft. This staying has been introduced into the principal Fairfield-built engines within the last few years.

We have stated that each set of engines of the new Cunarders has five cylinders, the high-pressure over the low-pressure, connected with the ordinary sleeve stuffing box with metal packing, while the intermediate is in the center. A piston valve is fitted to each of the high-pressure cylinders, Buckley's packing being used in the valves. A slide valve is fitted to the interme-

diate and to each of the low-pressure engines, the valve gear being of the usual double-eccentric and link-motion type. The bearing surfaces in all cases are large. It is scarcely necessary to state that all the cylinders are steam jacketed, and are fitted with automatic steam traps for the return of the water to the hot well.

The condenser is rectangular in section, and fitted with ¾-inch brass tubes. It is built up in three parts, and it is divided into two portions, so that each low-pressure engine has its own condenser. At the back of the condenser there are two air pumps, each driven by a lever from each of the crossheads of the fore and after engines. From the crosshead also are worked pumps for lifting the water from the condenser to the feed heater, to which we shall refer later.

The circulating pumps and engines were manufactured by Messrs. W. H. Allen & Co., Lambeth. For the *Campania* and *Lucania* four pumps have been supplied for each ship. It has been usually the custom to construct circulating-pump engines sufficiently powerful to meet the maximum requirements for pumping the bilge, requiring a considerably larger engine than is necessary for the ordinary work of circulating, so that the efficiency relative to the work required under normal conditions is low. The engines for the *Campania* and *Lucania*, however, are constructed in such a manner that they will be equal to the work of circulating while working on the compound principle with steam at the full pressure, exhausting into the low-pressure steam chest or condenser at the option of the engineer, and at the same time with such an arrangement of the cylinders that by a simple hand-slide valve both cylinders may receive the high-pressure steam whilst pumping from the bilge, and therefore working to suit the maximum requirements.

By this method economy is effected while circulating, and the full power can be obtained for bilge pumping, the size and weight of the engines being at the same time considerably reduced. It is the first time that the system has been adopted for work of this kind, but it seems likely that it will be largely adopted in future. The engines are of a substantial pattern, the size of the

main crankshaft having been kept up in order to withstand any possible strain that might come upon the engines while doing the work of bilging. The speed of the pumps when circulating is 100 revolutions per minute, and when bilging 250 revolutions. An auxiliary condenser is provided in each engine room for the air and circulating pumps, and for use when the vessel is in port and the main engines lying idle.

Returning to the main engines, it may be noted first that the high-pressure pistons are fitted with Ramsbottom rings, while the intermediate and low-pressure pistons have each one ring fitted with Downie's packing. The piston and connecting-rods are made of steel ingots, that used for each of the connecting-rods weighing, rough, 25 tons, this weight being reduced to 10 tons.

The whole of the shafting throughout the two ships has been supplied by Messrs. Vickers, Sons & Co., Limited, Sheffield. The crank-shaft is 26 inches in diameter, and each of the three interchangeable parts weighs 27 tons, so that with the thrust shaft, which is 14 feet long, the weight of each crank-shaft is about 110 tons. Each thrust bearing has 14 rings of the ordinary horseshoe type. These bearings are in recesses at the after end of the engine room bulkhead, but are tied direct to the foundations of the engines. The propeller shaft is 24 inches in diameter, and is fitted in about 24-feet lengths, each length having two bearings. The pillow blocks are of cast iron lined with white metal. With the bossing out of the stern, as described in dealing with the building of the hull, it has been rendered possible to dispense with the usual outside shafting, and this enables the propeller shaft to be carried in the stern tube, as in the ordinary single-screw steamer. The stern bushes are of the usual design—brass lined with lignum vitæ, the shaft being sheathed at the bearing to suit. The propeller boss is also of Vickers steel, and the three blades are of manganese bronze cast at Fairfield. Each blade weighs about 8 tons. At the after end of the propeller shaft there is fitted one of Dunlop's governors, while governors have also been fitted to the main engines.

Another safeguard is provided in the form of an emergency gear fitted in connection with Brown's steam and hydraulic starting and reversing gear. This emergency gear is such that the engines are stopped automatically if a predetermined rate of speed is exceeded. The starting engine is of a well-known type introduced some twenty years ago by Messrs. Brown Brothers, and of which over 1,000 have been made. The engine consists of a steam cylinder in which a piston and rod by one stroke handles the links, the movement being controlled by a water cylinder with automatic valve gear. The novelty in connection with this gear is the addition of a governor, by the operation of which, when a shaft breaks or dangerous racing takes place, the links will immediately be shifted into mid-gear.

It is well known that any attempt to control the speed of triple-expansion engines without operating on the distribution of steam in each cylinder has up to the present not been satisfactory. With the ordinary governor, therefore, valves on each cylinder are necessary, and considerable complication is the result. It has, therefore, been considered much better to have marine engines strong enough to race up to, say, 50 to 100 per cent. over their normal speed, and after that to stop them or bring them to dead slow. The engineer's attention is thus at once called, and proper steps can be taken.

There are twelve large boilers, and two others for auxiliary purposes. The total number of furnaces is 102. The twelve main boilers are double-ended, 18 feet in diameter and 17 feet long, each having eight of Fox's corrugated furnaces, four at each end. There is a combustion chamber common to each pair of furnaces. One of the smaller boilers, 18 feet in diameter by 11 feet long, with four furnaces, is for the auxiliary machinery in the ship, while the smallest, 10 feet in diameter by 10 feet long, with two furnaces, is for donkey purposes when the vessel is in port, but all may be used in the propelling of the ship if considered desirable.

The boilers are fitted longitudinally, three in a row, and are placed in two groups in two water-tight compartments, separated from each other by a large coal bunker occupying the full width

of the ship, and 65 feet of its length. There are also coal bunkers over the boiler spaces, the floor of the bunkers curving to the sides in much the same way as the usual protective deck in cruisers. The fuel is passed on to the stoking floor by chutes at the side, fed from the main coal bunker between the boiler compartments. The total coal-carrying capacity of the bunkers is great, so that the vessels will, when employed as armed cruisers, be able to keep the sea for long periods, and be the more efficient for patrol duty. The bunkers, it may be stated, are coated with Messrs. Wailes, Dove & Co.'s bitumistic enamels. The two auxiliary boilers are placed forward. The main boilers, it may be remarked, are the largest yet made for the pressure—165 pounds. The plates for the *Campania's* set were supplied by the Steel Company of Scotland, and for the *Lucania's* from the Parkhead Works, and it may be worth noting that some of these plates measured 20 feet long by 7 feet broad, the thickness being $1\frac{17}{32}$ inches. David Cockburn's safety valves are fitted to each boiler.

For each set of boilers there is a funnel, which has a double casing. The inside diameter exceeds 19 feet, while the top is 130 feet from the bottom of the ship. In view of the space occupied by the boiler spaces and coal bunkers, the funnels are a great distance apart—about 130 feet.

The Fairfield Company have not in any of their fast mercantile steamers gone in for forced draft as a means of adding to speed. Nor is this due to lack of experience or of complete acquaintance with the system and its application, for one has but to recall the signal success of the trials of the first-class cruisers *Edgar* and *Hawke*, the engines and boilers of which were constructed at Fairfield,* to show that, even with relatively high air pressure in the stokehold, the boilers were found to work very satisfactorily and without defect being subsequently disclosed, as has been too frequently the case. The Fairfield Company, however, as well as their clients, the Cunard Company, prefer to get their results without such aid, and the stokeholds of the *Campania* and *Lucania* have, therefore, been left open.

*See "Engineering," vol. LIII, pages 12, 75, 80 and 330.

The boilers, occupying as they do the central part of the ship, would have taken up the best location for the public and principal state rooms if the usual air hatches had been left, and to compensate for the partial closing up of these to add to the passenger accommodation on the upper deck, immense ventilators have been provided by Messrs. Mechan & Sons, Glasgow, supplemented by fans, mechanically driven, for use when necessary for efficient ventilation. The cowls revolve on wheels round a circular track moved by an endless chain winding round a drum which is turned by a lever. One man may therefore do this work, and here it may be stated that a deck hand will be told off for this duty, thus obviating the necessity and inconvenience of firemen going on deck for the purpose. This natural system of ventilation will be a great advantage to the men working in the stokeholds.

The whole of the fans for use in the stokeholds of the two ships were constructed by Messrs. W. H. Allen & Co., Lambeth, and are driven by their new patent triple-expansion single-acting engines having one crank. There are twelve of these engines and fans on board each ship. The revolving parts work in a bath of oil in the usual way, and are constructed with extra large wearing surfaces. Messrs. Allen have constructed a number of these engines to comply with stringent conditions for running continuously without attendance, and these, we believe, have given considerable satisfaction.

The feed arrangements on the *Campania* and *Lucania* have been most carefully worked out, and, as in all other parts of the machinery, everything is in duplicate, to provide against contingencies. It is probably one of the most complete installations of Messrs. Weir's specialties yet placed in any vessel, comprising as it does feed heater, main feed, auxiliary feed, ballast, and general service pumps, and also evaporators, and it may, therefore, be described at some length. The economy and efficiency of this system of feed heating is now recognized, and the *rationale* of the system may be briefly stated thus: The quantity of steam condensed by the feed water represents an amount of steam which has been worked in a theoretically per-

fect engine, because after doing work the whole remaining heat, both latent and specific, is returned to the boiler.

The cold feed water is discharged to the heater* by the main engine pumps, and, passing through the spring-loaded valve on the cover of the heater in a thin sheet, is at once heated by contact with the steam taken from the intermediate cylinder exhaust, entering by the non-return valve on the side of the apparatus. A circular ring and conical spray-piece, with perforations, are fitted to mix the water and steam more uniformly. The water falls to the bottom of the heater, in which is placed a round galvanized iron float, with water-tight bottom and sides, but open on the top. This is suspended on two levers so as to move up and down with a parallel motion; the top lever spindle is carried through the door at one end, and is balanced by a lever and weight. The float is always full of water, and the weight is adjusted to balance it when one-half is immersed in water. To the weight lever another lever is attached, which actuates the throttle valve and controls the supply of steam to the feed pumps. When the water in the heater rises, the float is raised and the throttle valve opened, and when the water level is lowered the float follows and the valve is closed; the level of the water is thus kept constant in the heater, and the pumps are completely filled with water and prevented from drawing air. The pressure on the water in the heater is always considerably less than before it passes through the inlet valve, and its temperature is almost instantly raised by contact to that of the steam. The effect of these two simultaneous actions is to cause the feed water to give up the air and gases in solution, and these are removed, either to the atmosphere or condenser, through the small cock on the top of the air vessel on the cover. The water is now at the boiling temperature due to the pressue in the heater, is freed from air, and is thus without corrosive effect on the boilers.

From the heater the feed water is drawn by the main feed pumps and discharged to the boilers. Two pairs of main feed pumps have been fitted. Each pump, when working at a moderate speed, is of such power that one pair of pumps is capable

* For cut of Weir's Feed Water Heater, see JOURNAL, Vol. IV, p. 399.

of easily feeding all the boilers. These pumps have probably the most difficult task of any portion of the auxiliary machinery, as the average pressure on the pistons will probably be about 160 pounds per square inch, and the temperature of the water to be pumped will be over 200 degrees Fahrenheit. Steam to these pumps is regulated by the valve controlled by the float in the heater, thus making the pumps automatic in their action. Noticeable features of these pumps are the steam valve, the suction and discharge valves. The steam valve consists of a main slide valve and an auxiliary slide valve. The main slide valve is made of a circular form, and is moved by steam. The auxiliary slide valve derives its motion from that of the piston-rod, and its work is to distribute the steam to the two ends of the main slide valve. There is also an arrangement whereby the steam can be cut off at about three-quarters stroke, and cause the piston to slow down to the end of the stroke, thus allowing the valves to close quietly and avoiding shock. The suction and discharge valves consist of a number of small valves in one seat, providing a large water area with a small lift. The small valves are of special design, and are milled out of cold-rolled manganese bronze bars, to insure durability under the high pressure. The water ends of all the pumps are of gun metal, and represent the latest and most approved practice.

The features enumerated are common to all the pumps of the Weir installation, as in the case of the auxiliary feed pump, of which two have been fitted. These are nearly similar in size to the main feed pumps, and are constructed to draw from the hot well, sea, bilge and boilers. They discharge to the boilers or overboard, and are also arranged to constitute fire engines of a powerful type. Four auxiliary and general pumps have been fitted, having each about half the capacity of the main feed pumps. One of these is arranged horizontally on account of space requirements, and embodies the leading features of the vertical type. Two ballast pumps have also been supplied. Each of these pumps is capable of dealing with 140 tons of water per hour.

It is now scarcely necessary to enumerate the various advan-

tages to be derived from the use of evaporators, as these have for some time past been recognized as indispensable adjuncts of every well-equipped marine engine. Messrs. Weir have furnished to each of the new Cunarders four of their well-known type of evaporator, each capable of producing 30 tons of fresh water per day. It consists of a cylindrical shell, in which the heating surface is composed of a series of U-shaped solid-drawn copper tubes expanded into a tube plate and placed in the lower portion of the shell. The method of making the tubes uniformly efficient is by means of a specially patented arrangement known as the "contracted ends and return tube," which gives this type of evaporator its high efficiency. The same firm have also supplied a couple of their small fly-wheel donkey pumps, Admiralty type, for pumping the fresh water from the tanks up to the pantry, &c. Pumps of the Navy type have also been supplied by Messrs. Broadfoot & Sons, Glasgow.

It is an easy transition from the feed arrangements of the boilers to the steam-pipe connection with the engines. The whole of the main steam pipes in the two ships are wrought iron with lap-welded joints. These are the first Atlantic vessels fitted with these pipes, and the departure from copper has been determined upon after very careful tests carried out by Mr. Laing, which tests satisfied him that the wrought-iron pipes used gave greater security against mishaps. We published the results of these tests in a previous volume, but it may be here stated that a pipe of 11¼ inches bore, with metal of $\frac{3}{32}$ inch thickness, was subjected to hydraulic pressure up to 800 pounds to the square inch without showing any effect, and it did not burst until the pressure reached 3,100 pounds to the square inch. A much smaller pipe of 8 inches bore and of metal $\frac{5}{16}$ inch thick showed no movement nor leakage at 1,600 pounds pressure, and only burst at 2,800 pounds. The compression and tensile tests gave results equally satisfactory. The pipes used in the Cunard steamers were all supplied by Messrs. Stewart and Clydesdale, Limited, Glasgow. The largest pipes are of 20-inch bore, and the thickness of metal is $\frac{9}{16}$ inch. The large double-stop valve is placed conveniently for both engines, passing as it does through

the fore-and-aft bulkhead dividing the two engines. The steam pipes between the valve-chests, being wrought iron, had to be straight. They are in short lengths, and expansion and contraction are provided for by means of stuffing-boxes at the joints.

The electric installations in these ships have been carried out by Messrs. Siemens Bros. & Co., Limited, London. The electric generating plant is in duplicate, so that in no case can there be an entire collapse, two sets being placed each side of the center-line water-tight bulkhead. There are in each ship about 1,350 lights, and these require an output on the part of the dynamos of 42,000 watts; the current is distributed throughout the ship by about 50 miles of wire, thickly insulated in vulcanized india-rubber, and laid on the return-conductor system in connection with Siemens' patent distributing boxes. The whole lamps in each ship would give a light equal to about 22,000 candles, absorbing 135 horse power.

All the engine and boiler-room fittings, and those for exposed parts, are water-tight, the glass globes bedding on rubber rings, and being held in position by metal straps. There are several portable lights, provided with flexible wires. In the engine room oil has been a cause of trouble in the destruction of the indiarubber insulation of the wires, but here the flexible wires for the portable lamps in the machinery spaces have all been insulated with cotton impregnated with tallow.

The electric current is generated in each ship by four dynamos, each driven by a separate engine. All four are similar. Two are placed in the recess in the port engine-room bulkhead, formed primarily for the accommodation of the thrust shaft, and two in the same position in the starboard engine room. Any two are equal to running the 1,350 16-candle power lamps in the ship, including the large reflectors of eight lights each for working the cargo, together with the search-light for facilitating the navigation of the ship into port—picking up the moorings—and for scout purposes should the vessel be engaged as an armed cruiser.

The engines for driving the dynamos are of the open, vertical, compound "Crescent" type, manufactured by Messrs. G. E. Belliss & Co., Birmingham. Each engine is designed for an output

of 70 brake horse power when running at 275 revolutions per minute, and with 100 pounds steam pressure at the engine stop valve. The cylinders are 9½ inches and 15 inches in diameter by 10 inches stroke, and are supported by strong cast-iron back standards, which also form the guides for the piston-rod cross-head shoe, and from the bed plate in front by two stout steel forged columns. The principal feature in these engines is the patent single slide-valve placed centrally between the two cylinders, which does away with a complete set of valve gear, eccentric rods, &c., used in ordinary double-cylinder engines. It is of the balanced-piston type, and admits and exhausts the steam from both cylinders. The governor balls are controlled by springs, and the gear is arranged to operate the single eccentric directly. As there is only only one slide valve, the powers in the high-pressure and low-pressure cylinders are maintained practically equal at all loads, this being unattainable when the governor acts only on the high-pressure slide valve. These engines are guaranteed to have a variation of speed between full and no load of not more than 2 per cent. The cranks are placed opposite each other, and the pistons are equally weighted, thereby making the engine perfectly balanced. The engines have been designed for long continuous running at full power, having specially large bearing surfaces, all pins being case hardened and working in phosphor-bronze bearings. The lubrication throughout is very thorough, the bearings all being supplied with oil from the three sight-feed oil boxes placed on the cylinder front.

The dynamos are of Siemens' well-known H.B. type, having drum armatures of 15 inches diameter. Each dynamo is capable of giving an output of 420 ampères under 100 volts pressure at 275 revolutions, and thus can run 700 sixteen-candle power lights. The magnets on the dynamos are supported on brass feet bolted to the engine-base plate, which is extended for the purpose. The armature shaft is coupled by an ordinary flange coupling to the shaft of the Belliss engine.

The most important item in the navigating appliances of the ship is clearly the steering gear, which has been provided by Messrs. Brown, Bros. & Co., Edinburgh.

The general principle of this gear, now well known, is that the steering engine is attached to the rudder-head without the intervention of chains and ropes, which frequently get out of gear; that it lets go the rudder when unduly strained, allowing the rudder to return automatically to its former position when the abnormal strain is gone. The general principles and details were explained by Mr. A. Betts Brown, the patentee, in a paper read some years ago before the Institution of Naval Architects. The rudder-head is turned by means of a tiller having at its after end a double jaw fitted with bearings, and carrying between the jaws a pinion running in a horizontal plane and engaging with a toothed quadrant bolted to the deck. The steering engine is, as stated, carried on a tiller and moved round with it, receiving and exhausting steam through a double stuffing-box arrangement mounted on the axis of the tiller. The cylinders have piston valves, and the reversing is effected by changing the direction of the steam entering the cylinders through a slide valve placed between the latter. The arrangement has been considerably modified to suit the limited space at disposal consequent on the fine lines of the new Cunarders and the fact that all the gear had to be below water-line, a condition insisted upon by the Admiralty for vessels which, as in the case of the *Campania* and *Lucania*, are destined to act as armed cruisers in times of war. Both the main and auxiliary steering gear are worked by steam, there being no hand gear. The space at the disposal of the designer was but 6 feet wide at the rudder-head, gradually increasing with the curved lines of the ship to 22½ feet wide at the center line of the auxiliary gear, the length being 44 feet. Owing to the fineness of the after part of the ship, it was decided to fit connecting rods from the rudder crosshead to the steam tiller, which in this case works forward instead of aft, as in some previous arrangements. These rods, being about 20 feet long, enable the main steering gear to be placed in such a position as to work clear of the ship's frames. The tiller, which is of cast steel, is about 17 feet long and acts upon a 4-foot rudder crosshead. All the machinery involved is a bronze worm-wheel, connected, by an internally expanding friction clutch with spring

relieving gear, to a steel pinion about 18 inches in diameter working in the cast-steel segmental curved rack of about 13 feet radius on the pitch line. The steering engines for driving the bronze wormwheel work in an inclosed oil-tight tank on the end of the tiller, having two valveless oil pumps, which throw a constant stream of oil over the working parts, the charge of oil being renewed about once in three months. The power of these engines and gearing is sufficient to strain the rudder-head to one-half its elastic limit.

In a ship of such size and speed as the *Campania* it was considered that any reasonable number of hand-steering wheels would not render it feasible to steer the ship when the steam steering gear was deranged, therefore auxiliary steam steering gear is fitted forward of the main steam tiller. The auxiliary engines are of one-third the power of the main engines, but are geared up by cast-steel gearing, with pitchwheel and chain, which latter lays hold of the end of the tiller, so that the ship can be steered as efficiently with the auxiliary as with the main gear. The auxiliary gear is connected and disconnected by means of an internally expanding clutch in the same way as the steam tiller, so that two reliable brakes are always ready to be applied to the rudder in case of an accident, and the engagement of either gear can be effected within a minute without involving the shipping of bolts or clutches. The efficiency of both brakes depends, of course, on the presumption that the steam tiller does not break, such an accident being a most remote possibility, as the latter is of a strength largely in excess of the resistence to torsion of the rudder head.

To guard against the possibility even of this accident, however, a pair of powerful hydraulic cylinders are provided, the rams of which lay hold of the connecting-rod pins at the steam tiller end. Water circulates between these cylinders, passing through a communicating valve, so that the attendant could at once shut the valve and arrest the motion of the rudder should such an extremely remote contingency occur as the breaking of the tiller. The side connecting-rods are also of such a section that one would be sufficient to steer the ship with perfect safety,

and all the fittings connected therewith are sufficiently strong to resist the unbalanced strain due to steering with one rod only.

The valve gears on the main and auxiliary steering engines are operated upon by a hydraulic telemotor cylinder connected by two lines of ½-inch diameter copper pipes to the bridge, where the telemotor is placed. In addition to this steering station there is also one on the poop, which communicates by means of a vertical shaft with the valve gear. Finally, in the event of the ship being used as a cruiser, and the telemotor pipes being shot away, as well as the after steering station, there is an under-water steering station from which the vessel can be steered with the aid of telegraphic instruction from the bridge. It will be seen, therefore, that every contingency has been guarded against.

The crew, all told, numbers 415. It is divided into three groups—sailing, engineers', and stewards' departments—as follows:

Sailing Department.	Engineers' Department.	Stewards' Department.
Commander.	1 chief engineer.	1 chief steward.
6 officers.	21 engineers.	105 stewards.
Purser.	2 refrigerating engineers.	45 cooks, bakers, &c.
Surgeon.	1 dock engineer.	8 stewardesses.
Carpenter and joiner.	8 electricians.	
Boatswain and 2 mates.	2 storekeepers.	
6 quartermasters.	1 "donkey" man.	
1 lamp-lighter.	18 oilers.	
40 seamen.	9 leading firemen.	
	75 firemen.	
	57 trimmers.	

In sailing department	61
In engineers' department	195
In stewards' department	159
Total crew	415

SPEED TRIALS OF THE *CAMPANIA*.

It was said by Sir John Burns, at a cruise of the *Campania* on Saturday last (April 15th), that he did not think he could pay a higher compliment to the Fairfield Company than to state that if the *Campania* and *Lucania*, with their vast power and

great size, could attain the position the *Umbria* and *Etruria* gained in their time and generation, the Cunard Company would be entirely satisfied. That this is probable the speed trials of the *Campania* have demonstrated; and these trials, it should be noted, did not consist of the ordinary arrangement of runs on the measured mile, when it is usually somebody's business to see that the engines and boilers are for the moment under the best conditions. On the completion of the vessel at Glasgow she sailed down the Clyde, and a long series of preliminary trials was commenced, the engines being steadly worked up to full power. One of these runs included a voyage round Ireland to Liverpool, where she was docked with the view of having her hull cleaned, for she had lain in the Clyde for several months. This work completed, she returned to the Clyde to undergo the stipulated power and speed trials. Arriving there, the vessel proceeded on runs at full power, the final trial taking place on Thursday, April 13th, when, as at some of the other official trials, Sir W. G. Pearce and the directors of the Fairfield Company were on board, while the Cunard Company was represented by Sir John Burns and Sir William Forwood. The results of several long-distance runs with the vessel in sea-going trim, and including one or two trips past Ailsa Craig, may be thus summarized:

Steam pressure in boilers	165 pounds.
Vacuum	28 inches.
Mean indicated horse power (both engines)	31,050.
Average revolutions of engines	84.
Speed (mean)	23 18 knots.
Mean draught of vessel	25 feet.
Displacement, about	18,000 tons.

The run was made within known distances, so that the speed could accurately be determined without the use of logs. The coal used on the occasion was Nimmo's Slamannan.

The boilers gave an ample supply of steam, and even the fans provided for ventilating the stokehold were not used. These, indeed, have only been run to comply with their own tests. The temperature of the stokehold was most satisfactory. Of this the

writer can speak from experience. The funnels are of such height that the draft in the furnaces was ample, while the stokehold ventilators have immense cowls. One feature of the engine performance was the remarkable uniformity of speed, the pistons running about 1,000 feet per minute, which is by no means excessive now-a-days, so that, although the speed of the vessel exceeds by a nautical mile per hour that guaranteed, there is prospect of even higher results being attained on the Atlantic. The results were certainly not got by straining the machinery to the greatest limit.

The vibration of the ship is of great interest, alike to the technical reader in view of the paper by Mr. Schlick published last week, as well as to the general reader, since it affects the comfort of the traveling public. It may be noted that Mr. Schlick's investigations lead to the assumption that severe vibration is due to the synchronizing of the vibration of the ship with the revolutions of engines, and that since the period of vibration of the ship depends in large measure on the length or size of the ship, the effect of longer vessels is to make the period of vibration longer, while, with increasing speed, the number of revolutions of the engines increases, so that with great ships there may be no vibration. It would have been interesting to have had accurate results got with Mr. Schlick's pallograph at the various speeds of the *Campania;* or to have had on board Mr. W. H. White's young man from the Admiralty, whom he described as a human pallograph, since, owing to his excessive physical sensitiveness, he could determine accurately the extent of vibration without an instrument. The present writer, fortunately or unfortunately, is not quite so sensitive, so that he can only give the general results. When the vessel was started there was no vibration; as the speed slowly increased, the vibration became more and more perceptible until three-quarter speed was reached, but even then it was only slight, and was chiefly at the extremity of the vessel, while when the engines were working at 83 revolutions and the speed of vessel was 23 knots, it was almost impossible to determine that the vessel was moving. When lunch was being

served to a large company of ladies and gentlemen in the dining saloon on Saturday last, many were suddenly startled by great throbs, so very decided that the number could be easily counted at 82 per minute. This was outside Ailsa Craig. It was so regular, however, that one easily got used to it, but it as suddenly ceased. The explanation was that the great rudder, with its area of 253 square feet, was thrown over to turn the vessel. And this rudder it may be stated, turns the vessel within her own length. The absence of vibration or of "working" of the ship is due in a large measure to the great strength and the careful designing of all detail, a work which requires experience in all types of marine construction and involves constant attention; but the managers at Fairfield have that experience which enables new conditions to be met with satisfactory results.

Altogether, in respect of both speed and comfort it may be regarded as certain that the *Campania* and *Lucania* will be improvements on the *Umbria* and *Etruria* commensurate with the advance of the times, and will thus meet Sir John Burns' requirements. These latter vessels "in their time and generation" reduced the outward record by 9 hours and the homeward record by 7½ hours. If the *Campania* maintains 23 knots on the Atlantic, she will cover the 2,800 miles which separate Roche's Point from Sandy Hook in just over 5 days; but even at a lesser speed she will take many hours off the duration of the Atlantic voyage.

The first Atlantic record-breaking vessel built by the Fairfield Company was the *Arizona*, and since that time they have constructed 16 Atlantic vessels. The series includes the *Arizona*, *Alaska*, *Oregon*, *Umbria* and *Etruria*, and the two vessels for this summer's traffic, the advent of which, with their respective compeers, will probably result in the Atlantic voyage being done in 2½ days' less time than in 1879. Of course other firms besides Fairfield have contributed to this result; but the *Campania* and *Lucania* are in advance of all other craft as regards speed, while they are second to none as regards safety, for this was the first consideration, it being the proud boast of the Cunard Company

that in their transatlantic traffic they have never lost a life. We have enlarged upon this point in our description of the ships, showing that the vessels are subdivided by transverse bulkheads, and that all engines and machinery are in duplicate. Indeed, a thousand and one evidences might be adduced to show that in working out the details of construction and design the requirements of safety have been carefully considered.

The results establish the fact, if ever doubt was entertained, that Fairfield still maintains that prominent place which for years she has taken in marine practice, and that the present managers of the technical departments, Messrs. White and Laing, are possessed of qualities similar to those whereby John Elder, Dr. Kirk, Sir William Pearce and Bryce-Douglas, each in turn, raised the establishment to its eminent position.

The following account of the first transatlantic voyage of the *Campania* is from the New York "Sun" of April 30th:

FIRST VOYAGE OF THE *CAMPANIA*.

The Cunarder *Campania* dropped anchor last night in the upper bay, south of Bedloe's Island, after an eventless run of 6 days 8 hours and 34 minutes from Queenstown to Sandy Hook.

Her engines worked with perfect smoothness, and she was not slowed down a moment, except for fog. She was run, however, Chief Engineer Payton declares, at about three-quarters speed, except on Thursday last.

Then, despite the high head sea and southwesterly gale, she reeled off 503 knots for the nautical day ending when the sun was at zenith.

The chief engineer may have been that day desirous of breaking the best day's record for maiden voyages. He did it, whether he desired to or not.

He also succeeded, apparently without effort, in eclipsing all maiden voyages from Queenstown. The maiden voyages of the three swiftest liners are:

City of Paris, 6 days 18 hours and 53 minutes.

Teutonic, 6 days 14 hours and 45 minutes.

Majestic, 6 days 10 hours and 30 minutes.

This splendid initial voyage of the *Campania* was in unusually unfavorable weather. She passed out of Queenstown harbor on Sunday afternoon (April 23d) and met moderate head winds until Thursday.

Then there was a fresh gale from the west that combed up tall seas. On Friday it was blowing a whole gale, with hurricane squalls, and she made only 426 knots. She ran at reduced speed through a dense fog off the banks.

She covered a long southerly course of 2,869 knots in daily runs of 475, 467, 483, 503, 426, 450, and 95 knots. Her engineers are confident that she will make the record of the *Paris*, 5 days 14 hours and 24 minutes, a mere memory before the end of the coming summer.

It had been predicted with some degree of confidence by shipping men who recalled that the big ship had on her trial trip developed a speed of 23.50 knots, that she would beat the *Paris* of the American line into port. But she didn't.

The *Paris*, over a course of 3,127 knots from the Needles, outside Southampton, developed an average speed of 19.3 knots. The *Campania's* average hourly speed for the voyage from Daunt's Rock to Sandy Hook was 18.60 knots.

In making this comparison it must be remembered that the British boat was speeded only one day in very unfavorable weather, and that the *Paris* was run for pretty nearly all she was worth.

Allowing for the enthusiasm of the engineers, who say that the *Campania* was run at not more than three-quarter speed, it may be said that she will, without undue exertion, make the trip from Daunt's Rock to Sandy Hook Lightship within five days and five hours, after her engineers have become familiar with her little ways.

The following account of the first trip from New York to Queenstown is from the New York daily papers:

LIVERPOOL, *May* 12.—The new Cunard Line steamship *Campania* which sailed from New York for Liverpool on May 6 arrived at Queenstown at half-past 9 o'clock this morning, having made the passage from Sandy Hook to Queenstown in 5 days 17 hours and 27 minutes, the quickest passage eastward yet made by any steamer.

The *Campania* landed her passengers here at 10 o'clock tonight, and thus again beat all previous records. She steamed from Queenstown to Liverpool in ten hours and three minutes, or at the rate of 22.48 knots an hour. The officers of the *Campania* cannot say enough in praise of her engines and her seagoing qualities.

The *Campania* has practically broken all records and is the fastest steamship afloat. Yesterday she ended a voyage on which she broke the records for the fastest eastward trip and for the largest number of knots made in a single day. If she had made the same average speed on her westward passage she would have beaten the westward record by over an hour. On her trip to New York she broke the maiden records for the fastest trip and for the greatest number of knots in a single day.

The big Cunarder and the American Line steamer *Paris* left New York last Saturday. The *Paris* started at 9.13 A. M. and the *Campania* swung out into the stream at 9.30 A. M. The time of ocean races is reckoned from Sandy Hook Lightship. The *Paris* passed the lightship at 10·58 A. M. and the *Campania* at 11·13 A. M. The *Paris* was sighted from Fire Island at 12.56 P. M. and the *Campania* at 1·25 P. M. The machinery of the *Campania* soon began to do effective work, and 109 miles from Fire Island the big Cunarder left the *Paris* far astern. The *Campania* passed Brow Head at 7 A. M. yesterday and arrived at Queenstown at 9·47 A. M. The passage of 5 days 17 hours and 27 minutes beats the record held by the American Line steamer *New York*, which was 5 days 19 hours and 57 minutes. The *Campania's* daily runs were 510, 490, 474, 517, 493 and 384 knots, the total distance covered being 2,868 knots. This was over the winter course. The record of the *New York* was made in August, 1892, upon a track of 2,814 knots. Traveling over the

track of the *New York* the *Campania* could have lowered the record two more notches. The engines of the *Campania* worked smoothly, the weather was fine and the water was fairly smooth. The average speed of the *Campania* was 21 knots an hour, which smashes all records for eastward passages. The *New York's* record is 20.11 knots an hour, and the *Fuerst Bismarck* made a record of 20.8 knots.

The most remarkable feature of the *Campania's* trip is her record of 517 knots in a nautical day of 23 hours and 3 minutes. On that day her average speed was over 22 knots an hour. The best previous record was made by the *Paris*, which made 530 knots on a Western passage in a nautical day of nearly 25 hours. If the *Campania* had been traveling Westward she would have steamed at least 550 knots under the same conditions. The record for the Westward passage, which is still held by the *Paris*, was made in 5 days 14 hours and 24 minutes, over a course of 2,782 knots. Had the *Campania* gone over the course of the *Paris* at the rate she went Eastward, she would have made the trip in 5 days 13 hours and 22 minutes. When the *Campania* returns to New York her owners confidently expect that she will break the record for the Western trip and dethrone the *Paris* as the queen of the seas.

MARINE BOILER FURNACES.

[Discussion of the paper by Mr. D. B. Morison, published in the February issue (No. 1, Vol. V) of the JOURNAL, pp. 11–38.]

Owing to the great interest that had been evinced in Mr. D. B. Morison's paper on "Marine Boiler Furnaces," a special meeting of the North-East Coast Institution of Engineers and Shipbuilders was held on January 30th in the Lecture Hall of the Literary and Philosophical Society, Newcastle-upon-Tyne. Mr. Wigham Richardson presided, and there was a good attendance.

Mr. J. T. Milton, the chief engineer surveyor to Lloyd's Register, who resumed the discussion on Mr. Morison's paper, said: "I have been pointedly asked by Professor Weighton and by Mr. Patterson for an explanation of the peculiarity of the diagram shown in figure 19, which has led them to suppose that Lloyd's Register unduly favors the Purves furnace by according to it a lower factor of safety than is required with other furnaces.

I stated at the last meeting that this was not the case, and that Lloyd's Register treated all furnaces with equal fairness so far as the experimental data at their disposal permitted, and the following facts, taken in conjunction with the information given in Mr. Morison's tables, will show the correctness of this statement. At the outset it is scarcely necessary to state that our knowledge of the strength of furnaces or any other structure must be based upon the results of experiments of some kind, and, further, that even with the most simple experiments apparently duplicate tests do not give identical results. Even the simple tensile test, the simplest of all experiments, will give differences of results both in ultimate tensile strength in elongation from pieces cut from side-by-side positions in the same plate. It can only, therefore, be expected that the results of experimental

tests of furnaces, when each one will be made from a different plate, will show somewhat different results.

Mr. Morison's tables show these differences. In the last column but one of each table, from I to to VII,* the results show (by Mr. Morison) that, even making all allowances for slight differences in diameter, thickness, and original strength of material, there are considerable differences in the results of similar tests. For instance, in Table II, in the last six tests there is a difference of 17 per cent. between the best and worst result; in Table III, the difference is 20 per cent.; in Table IV, it is 11 per cent.; in Table V there is about 70 per cent.; in Table VI there is 20 per cent.; and in Table VII there is 4 per cent. Now, each of the experiments in any one table are of the same reliability, and, therefore, the reasonable way of dealing with the information they give is to take the mean of the results. Now, if we turn to the diagram, figure 19, and the corresponding Table VIII, we see that this has not been done.

Commencing at the left, the spots 1 and 2, marked L, each represent two experiments out of the six recorded in Table II. In this table the variation is 17 per cent., but the two selected are practically the two best results. Spots 3 and 4 refer apparently to the first two and the last two of the experiments recorded in Table V. The last two of these are part of a series of six, including differences of results amounting to 70 per cent., and it is to be noted that these two are apparently practically the worst of the six. I may here say that the experiments in Table V are not known at Lloyd's Register, so that they can have no influence in determining the rules. Spot 5 apparently represents two experiments selected out of the seven recorded in Table VI. Here again it is to be noticed that one of these is the very worst of the series. Spot 6 represents the mean of two out of three experiments recorded in Table IV. Spot 7 represents a selection of two out of the five records given in Table III. The remaining spot represents two out of six experiments recorded in Table VII. Even if it were possible to fairly repre-

* The whole of the tables and diagrams are given in the February number of the JOURNAL.

sent the factors of safety required by Lloyd's Rules in the manner attempted in figure 19, it certainly, in my opinion, is misleading to give selected results instead of mean results, or at least, if selected, the basis of selection should be the same in all cases.

Turning next to Factor of Safety, Lloyd's Rules for such furnaces as are under discussion contain an expression $(T-2)$. The effect of this is seen in the diagram. Furnaces, like all other parts of boilers, are subjected to loss of strength through corrosion, and Lloyd's Rules have to provide in some measure for this.

The result of the expression $(T-2)$ is that as thin furnaces lose a greater proportion of their strength than thick ones from a given amount of corrosion, they are originally required to be made stronger. After an amount of corrosion represented by about one-eighth of an inch, the various thicknesses of furnaces, if proportioned by the rule, will still possess an equal margin of strength. If, for instance, a five-eighths furnace, when new, possesses a margin of five times, and a three-eighths furnace of six times, when each is weakened by, say, one-eighth of an inch of corrosion, each will have a margin of four times. It is seen from this that only furnaces of the same thickness can be illustrated by such a figure as that in figure 19.

Now, as to the actual figures by which Lloyd's Rules were determined for the different classes of furnaces. The experiments were those given by Mr. Morison in the last six tests recorded in Table II, the whole of those recorded in Tables III and IV, VI and VII. The exact figures in our records are not in all cases precisely as given by Mr. Morison, but in the main they very fairly agree. It will be convenient if we commence with the results of Purves' furnace, as recorded in Table VI. Of these experiments the first two were discarded because they were made of such thinness as could not be made in practice. (I believe the plates were planed out of thicker ones, as it was impossible to roll them.) The mean of the figures given in column 12 against the five tests is 74,915, and this divided by 1,160, the coefficient in Lloyd's Rules for Purves' furnaces, is 64.6. I

may here say that the records given by Mr. Morison of the tensile strength of the steel used in these furnaces is not in accordance with the information we have, which led us to take 27.2 tons per square inch as the mean strength of the five tests.

Next turn to the six last tests given in Table II, which are the tests on which Lloyd's Rules for Fox's furnaces are based. It is noticed at once that the tensile strength of the steel used is 29 tons mean. As this undoubtedly had an effect in making the furnaces stand a higher pressure in order to compare with the previous tests, the results must be reduced in the proportion of 27.2 to 29. Lloyd's Rules take the greatest diameter as the basis for this type of furnace, and give a coefficient of 1,259. Mr. Morison, in column 12, has taken the mean diameter. If his figures are made to agree with the greatest diameter which is taken in Lloyd's Rules, the mean of the column comes to 86,723. This divided by 1,259, and then reduced as above indicated to the proportion of 27.2 ÷ 29 gives 64.6, the same figures precisely as with Purves' furnace. The figures in Table III are not correct as regards diameters. Mr. Morison has taken as mean diameter the greatest diameter minus the thickness of plate, whereas the mean diameter is about two inches less than the greatest diameter. Calculating the figures, which ought to be in column 12, if the greatest diameter were taken, we arrive at a mean result of 57,851, which, divided by 912, the coefficient in Lloyd's Rules, gives 63.4, which is slightly different from the preceding. I should observe that we have no record of the strength of steel in this case; neither, apparently, has Mr. Morison, who merely states, "assumed 28 tons."

The next furnace to be considered is Holmes', the results being given in Table IV. The mean of the figures in column 12 gives 60,995, which, divided by 945, the coefficient in Lloyd's Rules, gives 64.55, which is practically the same as in Fox and Purves' furnace. The mean strength of steel is recorded as 27.2 tons, the same as in Purves' furnace.

This deals with all the furnaces except the Morison. In dealing with this, it is necessary to refer to the experiment on Fox's furnace. In each of the six tests on these, the usual 6-inch plain

portion was left at one end, and four out of the six furnaces failed at these ends; three out of the six did not yield in the corrugation at all. These results show that a 6-inch plain portion is likely to be slightly weaker than the corrugated part. In the Morison furnace there is the same amount of plain portion to be provided for. In the actual tests, however, the plain portion was only $4\frac{1}{2}$ inches and $\cdot 5$ inches long, and did not, therefore, so fairly represent actual practice as the Fox tests. Although these plain portions were shorter, two out of the six tests failed at the shorter flats. It could therefore only be considered, in comparing these furnaces with Fox's, that, the plain portions of 6 inches in length in each being the weakest portion of each, the rules for both forms should be identical.

I am grateful to the council for permitting me this opportunity of explanation, and I am sure that all the members of our Institution will feel with me that it is desirable that no undeserved stigma should be allowed to be attached to the Committee of Lloyd's Register, whose reputation for perfect fairness and impartiality is world-wide.

Mr. Geddes said that in the position he occupied once for several years, as the representative of a forge company, he had been frequently called upon to supervise the repairing of distorted furnaces, and he never had the slightest hesitation in applying fire to such work, except on one occasion, and that occasion altered his mind to some extent.

It was in connection with a plain furnace in an Atlantic passage boat, which had given way. Before taking the work in hand he talked the matter over with the consulting engineer of the company, and they decided to proceed with the work. They built a wood fire on the under and upper sides of the plate, and then with a pair of jacks they placed it in its original position. When it was finished they thought it was a good job, but before going away he went to the furnace to take a final look, and was surprised and mystified to find that a crack had opened in a longitudinal direction. The crack was at least six to seven inches away from the point that they had dealt with. He found

it difficult to come to a conclusion on the matter. He thought at first that perhaps it was due to defective or inferior material, and he made inquiries. He had not access to the records of the firm that made the plate, but he learned sufficient to show that the material was of high tensile quality—28 to 30 tons. It was a half-inch plate, he believed. He was fairly convinced in his mind that the crack was due to a disturbance on the steel between hot and cold water, and he did not think the plate would have cracked had it been of milder steel. Ever since that time he had been very careful to inquire into particulars regarding a furnace before he attempted to try it with heat.

It should never be forgotten that whatever the tensile value of the plate might be that was put into a boiler, it was considerably increased by coming into contact with the fierce heat of the fire. In contradistinction to the last case, he mentioned one connected with a celebrated liner where there was a serious collapse. A practical engineer of Liverpool calculated very judiciously that the plates were likely to be of high tensile value, and that therefore to put fire to them would be to run considerable risk. Instead, he tried to put the distortions together again, and this was done. But it required a pair of 100-ton jacks to put them right—these were appliances that could not be secured everywhere—and he contended that if the furnace had been of milder steel 20-ton jacks would have been sufficient.

He thought that mild steel should be used where plates were likely to be subject to high heat. It had been suggested that Mr. Fox, of the Leeds Forge Co., must have great confidence in high steel, otherwise he would not have gone in for it in the manufacture of his own furnace. He (Mr. Geddes) was in a position to say that Mr. Fox was not now, and never had been, an advocate for high tensile steel. That gentleman was compelled from commercial reasons to get fully abreast of a competitive furnace that had taken the initiative and had got a rung of the ladder above Mr. Fox. They had got this rung, and were able to dance to a very pleasant tune, but the shipowners had to pay the fiddlers.

Mr. John F. Walliker said that in Mr. Morison's paper, [page 15, February number], they had the words, "The material should

be so disposed as to give the greatest resistance to collapse without undue rigidity in a longitudinal direction." This appeared in the face of it a strong commendation of furnaces of the rigid type, but as they had had comparatively no trouble with plain furnaces from their form, which was rigidity itself, it followed that the only "breathing" required was in allowing room round the furnace.

There was also a note that the furnace should be of such formation that it could be easily scaled, &c. This, he thought, should be looked after more than it was—some engineers thinking they had done all that was required when they had put some peepholes over the tops of the furnace. If, however, instead of this they would take away the bottom row of tubes (generally full of soot), and thus allow more room for men (and not boys) to enter, the tubes would have a chance of a longer life. Small doors were incentives to lazinesss.

He thought they were indebted to Mr. Morison for pointing out the internal stress under ordinary circumstances, and to this, in his opinion, more than to material, they had to look for the reason of some of their failures.

Several of the speakers had expressed their disappointment at the information of the manufacturers about the suitability of what they called high tensile steel, but to his mind the latter speech of Mr. Gross cleared this matter pretty well up, and he thought if they would refer to this in the printed proceedings they would see it as he did. Mr. Gross stated that steels of from 26 to 30 tons were chemically identical, and he believed he was quite right. Let them take an ingot cut into two pieces. One half, say an inch thick, would break under the limit, but the other piece, if rolled into half thick, would only break at say 28 to 30 tons, yet they had identically the same chemical constituents.

His contentions, in conclusion, were: (1) that steel from 26 to 30 tons had for many years given great satisfaction in plain and patent furnaces; (2) that steel under these strengths had failed in the same degree as the higher tensile material; (3) that as to design—taking the lessons learned from careful experiments such

as those indicated by Mr. Milton—ample breathing space and proper handling of the material in the shop would in the future give them the immunity from accident they had had in the past.

Mr. Morison then replied. He said the value of an engineering paper of a practical nature depended to a great extent on the discussion, and he took the earliest opportunity of thanking the various gentlemen who had given their views on the very important subject of marine boiler furnaces. At no period in the history of marine engineering had furnace accidents been so numerous as of late, and he felt confident that the definite expressions of opinion from the many eminent engineers who had taken part in the discussion would result in an immediate and permanent improvement in the material of which furnaces were made.

It had been remarked, rather ungenerously, that the paper was written to advocate a certain design of furnace, but such was not the case, as he did his utmost to give each design equal consideration, and the descriptive matter relating to each furnace was obtained from, and submitted to, the various firms before being included in the paper. He was neither a steel maker nor a furnace manufacturer, and he claimed that the paper was written from the standpoint of a marine engineer. He did not propose in his reply to dwell on the merits or demerits of any particular furnace; his opinions were definitely expressed in the paper, and from what had been said he felt there was no cause to alter these opinions; he therefore left the question of the best furnace to be settled by practical experience in the future.

He was glad to hear Mr. Alex. Taylor so severely condemn the use of steel exceeding 26 tons tensile for furnaces, as he told them that this unsuitable material had caused untold annoyance and expense to all concerned. He quite agreed that shipowners' and boiler makers' experience of patent furnaces within the past few years had been unsatisfactory; he would go further and say that many furnaces at present in use that had been made of hard steel would crack eventually, more especially if they were in boilers having great variations of temperature, as in the case of forced draft.

His remarks as to corrosion at the firebar level seemed to have been misunderstood, as what was meant was that under exactly similar conditions it was his opinion that if corrosion did occur it would be more severe in a furnace of a rigid design, such as a plain furnace, than in an elastic design, such as a corrugated furnace; and also that variations of temperature had a bearing on such corrosion.

The desire on Mr. Gross' part to use higher and still higher tensile, for the sake of getting the Board of Trade and Lloyd's to grant higher and still higher coefficients, was after all only natural. As Mr. Gross said, there was really no chemical difference, and why not use the hardest steel the authorities would allow? But it was a striking fact that, with the exception of the engineer-in-chief of Lloyd's Register, who did not see why high tensile should not do as well as low tensile, no one who had spoken on the paper agreed with Mr. Gross as to the suitability of the high tensile steel we had been lately getting for furnace manufacture. The Leeds Forge Co., after an experience of fourteen years, and a manfacture of 30,000 furnaces, condemned it. Messrs. Schultz, Knaudt, of Essen, with an experience of twelve years in the manufacture of 16,500 furnaces, condemned it. The American furnace makers condemned it, and the leading Admiralties of the world condemned it, and yet Mr. Gross and Mr. Milton did not see why it shouln't do. He asked an eminent steel maker, not long ago, whether he would rather make high tensile steel or low. He replied, "Anybody can make high tensile steel, but it requires a steel maker to make low tensile; and, besides, high tensile admits of using inferior pig, which is impossible when dealing with low tensile." There was evidently more in this reply than appeared on the surface.

Prior to his appointment as general manager of the Leeds Forge Co., Mr. Gearing had been a furnace user and a furnace buyer, yet after becoming thoroughly acquainted with the details of furnace manufacture he was of opinion that high tensile steel was not the proper material to make furnaces from. Mr. Gearing had informed him that the Leeds Forge Co. had made 27,000 furnaces

of steel of from 23 to 26 tons tensile, and, although there were cases of collapse due to the excess of deposit, cracking was practically unknown, and it was since the use of higher tensile that this had formed a recognized accident. As the material in a furnace was subjected to more severe treatment than in any other part of a boiler, it was necessary that it should be subjected to the most severe tests, and yet it was a curious fact that Lloyd's, prior to May of last year, did not test furnace steel in any way whatever.

In reply to Mr. Milton, he must at once take exception to his remark that he indirectly accused him of unfair dealing with the factors of safety of the various furnaces, as no one appreciated more than himself the responsibility of Mr. Milton's position, and his evident desire to extend the same degree of consideration to all. It was possible for anyone, however, to make an error of judgment, and in his opinion the factor diagram was an illustration of such errors. The table of experiments on which the diagram was based was absolutely correct, as it was made under the direct supervision of the Board of Trade and Lloyd's officials. Consequently, it was not a question of opinion, but of fact, and he maintained that such difference as existed between the factor of safety granted to the Purves' furnace in 1887 and to the suspension furnace in 1891 was not only misleading to the users of furnaces, but grave in its effect on industrial undertakings.

Mr. Milton accused him of bringing forward many expressions of opinion which he hesitated to accept as proved facts. The greater portion of the paper, however, dealt with absolute facts, namely, the results of careful experiments, and such opinions as he had advanced were the results of practical experience. Mr. Milton was evidently inclined to favor the use of steel from 26 to 30 tons tensile as suitable for furnace manufacture, and, whilst being favorably impressed by Mr. Gross' views, would like some other steel maker to come forward and say which was right. Surely Mr. Milton did not propose to allow this most important question of suitability of material to remain until it was settled by the steel makers. Marine engineers had proved by experi-

ence that the furnaces made during the last few years were faulty, but it was not their province to deal with the question of chemical analysis, but rather to insist on obtaining a material that would not fail. He, for one, did propose to wait until the steel makers said which was right, but rather to use material which had been proved to be successful.

With regard to the Admiralty, Mr. Milton said they used mild steel, and had as much trouble with their furnaces as anybody else. He (the speaker) distinctly denied this, as whatever might have been the trouble with the Admiralty boilers, their practice with regard to the furnaces had been most successful, and, as an engine builder, he would rather adopt their specification for furnaces than any he had met with. The experience of the firm with which he was connected had extended over the use of about 1,400 corrugated and ribbed furnaces, and about 850 of these were made of steel under 26 tons tensile. Not one of the 850 was returned to the furnace makers. This was an experience similar to that of Messrs. Schultz, Knaudt, but in their case the number of furnaces dealt with was 16,500, and whilst the 16,000 low tensiles were most successful, the remaining 500 made of steel above 25 tons were such a distinct failure that the firm positively refused to make any more.

In concluding, Mr. Morison said that, if, in the course of his reply, there might be a semblance of curtness, he could assure them that it was not intended. His only object was to give the Institution as much information on the furnace question as he had been able to obtain, and if such information resulted in the slightest benefit to marine engineering, his desire in its preparation had been realized.

At the sixth general meeting of the session of the North-East Coast Institution of Engineers and Shipbuilders, **Mr. D. B. Morison** concluded his reply to the discussion on his paper on "Marine Boiler Furnaces." He said:

In replying to Mr. Milton's explanation of the factor diagram I again repudiate any intention whatever of attributing intentional unfairness to Lloyd's Registry, and I cordially endorse

Mr. Milton's remarks that the reputation of Lloyd's Registry for perfect fairness is world-wide.

In considering the strength of boiler furnaces, it is fortunate we have actual results of experiments as our base, as we thus deal with facts and not opinions. In the first place Mr. Milton points out the percentage of difference between the best and worst results of the various furnaces, and shows that this variation is as follows:

1892.	Fox experiments, .	17 per cent.
1888.	Farnley experiments, .	70 per cent.
1891.	Holmes experiments, .	11 per cent.
1886 and 1887.	Purves experiments, .	70 per cent.
1889.	Purves experiments, .	20 per cent.
1891.	Morison experiments, .	4 per cent.

This is graphically shown on diagram 18, and the noticeable fact is that the test results of the Morison furnace are the most uniform ever obtained, the difference between the best and the worst being only 4 per cent., or about seven times less than the mean variation of the entire series of tests of all the other types of furnaces.

Mr. Milton states that only furnaces of the same thickness can be illustrated by such a figure as the factor diagram, and that, if selected tests are taken, the basis of selection should be the same in all cases. I entirely agree with this; indeed, these were exactly my views, and, as I particularly wished to illustrate everyday practice, I adopted as my standard, or, as Mr. Milton would say, my "basis of selection," the thickness most nearly to that in actual use; consequently the factor diagram is calculated from the means of those test furnaces which are nearest to nine-sixteenths in thickness. These were as follows:

1890–'1.	Fox tests, ...	.5745
1892.	Fox tests, ...	.5745
1886.	Purves' tests, ...	.557
1887.	Purves' tests, ...	.612
1889.	Purves' tests, ...	.5815
1891.	Holmes' tests, ...	.536
1886.	Farnley tests, ...	.5535
1891.	Morison tests, ...	.5485

I much regret the way in which Mr. Milton refers to the *best* tests being taken in one case and the *worst* in others, in order to prepare the factor diagram, as it unfortunately implies dishonesty of motive. Mr. Milton justly remarks that it is undesirable that an undeserved stigma should be allowed to be attached to Lloyd's Registry. To this I would add that it is equally undesirable that any undeserved stigma should be allowed to be attached to my paper, and with your permission I will now remove it, as it entirely arises from a misconception on the part of Mr. Milton.

Referring to the factor diagram, Fig. 19, commencing on the left and dealing with the spots marked L only. Spots 1 and 2 (Table II), refer to the Board of Trade official tests of the Fox furnace, made at the Leeds forge in 1890–91. The tests were most exhaustive, and were made regardless of expense. Test strips were cut in the presence of the Board of Trade officer and sent direct to Messrs. Kirkaldy, of London, and after the results were approved by the engineer-in-chief, Mr. Traill, the six plates were made into furnaces, all the operations being conducted under the supervision of a Board of Trade officer, and finally the furnaces were tested by Mr. Samson, chief assistant to the engineer-in-chief. Lloyd's were present at these tests, and took a copy of the various results obtained by Mr. Samson. Table II is, therefore, correct in every particular.

The fifth and sixth tests being nearest to the average commercial thickness of nine-sixteenths, these were taken, and the mean of the two gives .5745. The variation of 17 per cent. between the best and the worst of the six tests is explained in my paper, as one of the five-sixteenth plates was rolled thin at one edge, and when made into a furnace this thin edge became a flat on the furnace end, and as the thickness of this thin plain end was .331, whilst the mean thickness of the body was .378, the furnace naturally collapsed at the thin plain end, and this explains much of the 17 per cent. variation which Mr. Milton refers to. The mean tensile of the two tests taken was 28.98 tons, and as the Board of Trade demanded, at the tests of the suspension furnaces, that the tensile should be between twenty-six and twenty-eight tons, I took twenty-seven tons as representing the latest official

requirements as my basis, and reduced the tensile of the two tests in question with their corresponding collapsing co-efficient to twenty-seven tons in direct proportion in exactly the same manner as Mr. Milton does—

$$29 : 27 = C : C'$$

My paper describes that the plain tubes are heated before being corrugated, and although they leave the rolls red hot, three out of the six were heated again for annealing purposes, and it is a noticeable fact that two out of the three furnaces which were annealed collapsed at the end flat and the other on the first corrugation, so that it would appear that the ends of the three furnaces which were annealed were well heated at that part and the tensile thereby reduced.

No. 3 spot, Table V, represents the result of the Board of Trade official tests on the Purves furnace in 1886. These tests were made with the same elaborate care as those previously described, the material being tested by Kirkaldy and the experiments conducted by Mr. Samson. Two furnaces only were tested, the mean thickness being .557, and as that thickness approached very nearly the nine-sixteenths basis, it may be inferred that the results represent the strength of the Purves furnace as manufactured for use in 1886. All the particulars of these two tests are therefore absolutely correct.

No. 4 spot, Table V, represents the mean of two out of six tests made on the Purves flue in 1887. The two selected for calculating the diagram were, as before, the nearest to the nine-sixteenths basis, viz., the fifth and sixth, which give a mean thickness of .612. These tests were carried out with the usual degree of excellence by the Board of Trade, and the whole of the results in Table V are absolutely correct. Mr. Milton observes that "it is to be noted that these two tests are practically the worst of the six," thus insinuating that they were chosen for that reason. A very little consideration would have shown Mr. Milton that his conclusions were entirely wrong and prevented him from making this unwarranted accusation. His remark simply draws attention to the fact, that by choosing as a basis the furnace having a thickness most nearly approaching that in general use, we get

with the Purves design a furnace of the lowest relative strength. I pointed out that physical fact in my paper and illustrated it very clearly in Fig. 18, a reference to which will show that the weakness is no mere accident of experiment, but a direct consequence of the reduction in the proportion of the material in the stiffening ribs to that in the plain part. It cannot be disputed, however, that the results of the two tests I took correspond to those that would be obtained from furnaces manufactured for use at that date, and that alone would justify my position. Mr. Milton says these 1886 and 1887 tests on the Purves flue were unknown at Lloyd's, but this does not alter the fact that they represent the true state of the manufacture at that date. One test only appears to have been made for Lloyd's in 1886, but in view of the Board of Trade tests made in 1887, its correctness is doubtful, and I did not include it in my paper.

Spot 5, Table VI, represents a mean of two of six tests made by the Board of Trade only on the Purves furnace in 1889, and, as before, I took the mean of the two furnaces, nominally nine-sixteenths thick, for calculating the diagram. It will be observed that the table is very incomplete, and, although I applied to the Board of Trade for particulars, I failed to obtain them. I also wrote to Messrs. John Brown & Co., and they very kindly sent me the tensiles of the test pieces cut from the plates from which the six furnaces were made, at the same time explaining that they could not give me the remaining particulars, as they had no record. Some time after I found these tensiles did not agree with the others I obtained, and I wrote Messrs. John Brown & Co. the second time, asking them to examine their records again, as I wished the table to be absolutely accurate. To this Mr. Gross kindly replied, confirming the tensiles before given, the mean of which is 28.25 tons. These tensiles are, therefore, correct, but we cannot tell how or where the furnaces collapsed, or whether the furnaces were heated for rounding up and again heated for annealing, or whether heated once only. In either case, however, the tensiles *correspond to those of the original plates*, as in all the other cases.

With reference to these tensiles, Mr. Milton says: "The rec-

ord given by Mr. Morison of the tensile strengths of the steel used in these furnaces is not in accordance with the information we have which led us to take 27.2 tons as the mean strength of the tests." It will be noticed Mr. Milton does not say 27.2 is correct, but that he was "led to take it," yet on this assumption the calculation regulating the manufacture of the furnace seems to have been based. It would be interesting to know whether the 27.2 represents the tensile of the furnace after annealing, but even if that is the case, it is still wrong, as the tensiles of all the other tests were taken from the plates after rolling and before they were made into furnaces; indeed, in the Board of Trade experiments no progress is made with the manufacture of the test furnaces until Mr. Traill approves of the tensiles of the test strips. The one test made by Lloyd's in 1890 being incomplete, and the tensile strength being unknown, I thought it better to make no assumption, as the six Board of Trade tests afforded all the data necessary for accuracy.

Spot 6, Table IV, represents two out of three tests made by the Board of Trade on Holmes' furnace. This table was revised by Messrs. Holmes & Co., and is correct. The tests taken were the second and third, which give a mean thickness of .536.

Spot 7, Table III, represents two out of five tests made by the Board of Trade on the Farnley furnace, and the two taken being the first and second as giving nearest the mean thickness of nine-sixteenths. The table was submitted to the Farnley Iron Co. and returned as correct. The diameters were the subject of special correspondence, and I am assured by the Farnley Co. that they are correct; consequently Mr. Milton would appear to be wrong. No record was kept of the tensiles, but the makers advised me that by taking 28 tons it would be approximately correct, and, as Mr. Milton also assumes the same, it in no way affects the argument.

Spot 8, Table VII, represents two out of six experiments made by the Board of Trade on the Morison furnace in 1892. The appendix contains a detailed report on these tests, the accuracy of which cannot be questioned, as every detail of manufacture was supervised by a special Board of Trade officer. The test

strips cut from the plates gave a mean tensile of 27.2, and it is worthy of special notice that these plates were twice heated after the test strips were cut, once before being corrugated and the second time after corrugation, this final annealing being watched by Mr. Samson, who made a special visit to the Leeds Forge for that purpose. Consequently, the tensiles of the furnaces as tested would undoubtedly be lower than 27.2 tons.

It will be observed that two out of the six collapsed at the end flat, as against three out of six in the Fox, and the length of the flat is practically 5 inches against 6 inches in Fox. This difference of 1 inch is made much of by Mr. Milton, although I was very careful to deal fully with the question of flats on furnace ends in my paper, giving a diagram of those test furnaces which had failed at the flats.

All engineers will agree that long flats on the back top end are undesirable, but if the furnace is made for removal on Ashlin's plan, or that adopted by Messrs. John Brown & Co., there need be practically no flat at that part, the radius of the saddle flange sweeping into the radius of the first corrugation, the design of the Suspension furnace being particularly suited to this arrangement, the long suspension curve being continued into that of the saddle flange, so that no abrupt cavity exists.

In furnaces which are not designed for removal it is necessary to leave about 2½ inches of the flat on the top back end, so as to make a joint if the furnace has to be cut out at any time. The next important flat is at the front top, but in no case in the Suspension furnace need this exceed 4¾ inches, as, when it does, the makers roll in a Fox corrugation. There remains, therefore, only the flat at the bottom back end, and even this is only about 6 inches long, and being below the line of fire and behind the bridge it does not accumulate scale, and therefore retains its original factor of safety.

Mr. Milton's contention is that if the Morison test furnaces had had the same length of flat as the Fox, viz., 6 inches instead of 5 inches, they would have given equal results, and goes on to say that a 6-inch flat is necessary for practical requirements. Let us assume for argument that a 6-inch flat *is* a necessary

standard. The question then arises, If the body of a furnace has practically the same resistance to collapse as a plain part at the end 6 inches in length, is there any benefit to be derived from making the body stronger?

My opinion is that, if the body of a furnace can be made stronger than the 6-inch plain part at the end, it is a stronger furnace as a whole, because the center of the furnace, or that part above the hottest part of the fire, is certainly the weakest in actual use and should be the strongest under cold water test. If we examine the Fox tests we find three furnaces collapsed on the flats and three on the body, thus it would be reasonable to assume that the body of a Fox furnace is equal in strength to a plain end 6 inches in length.

The question now is to determine whether the body of a Morison furnace is stronger than a 6-inch flat. It will be seen on reference to Fig. 23 that the mean length of flats of the three Fox tests which collapsed on the flats was six and three-sixteenths, and the mean collapsing co-efficient 80,850. Consequently, had the mean length of the flats on the Morison furnace been the same, the collapsing co-efficient would naturally also have been about the same. But the mean collapsing co-efficient (calculated on the outside diameter) of the 2d, 3d, 4th and 5th Morison tests, all of which failed on the body, is 89,537. Consequently, had the Suspension test furnaces had a length of 6 inches they would all have failed on the flat, and therefore the body of a Suspension furnace is proved by these tests to be stronger than a 6-inch flat on the end, and therefore stronger than the body of a Fox furnace.

Mr. Milton says nothing about the flats in the Farnley or the Purves, although the 1887 experiments of the latter show that a flat of $7\frac{1}{2}$ inches long was stronger in every case than the ribbed body, and still further, in the 1890 Purves' experiments, Mr. Milton admits that he knows nothing of the position of the collapse, whether on the flats or in the body, and yet he would seem to depreciate the value of the suspension tests. No statements or opinions, however, can alter the fact that they are the most uniform tests ever made, varying only (according to

Mr. Milton's own figures) 4 per cent. between the best and the worst, and also that, on an equal tensile, they gave the highest resistance to collapse on record.

The general accuracy of all the tables having been proved, I will now show by an illustrative example how the Lloyd's spots were calculated. Take Table II for this purpose, and select the two tests nominally nine-sixteenths thick, viz., the 5th and 6th. The dimensions are—thickness .574 and .575, diameter over corrugations 35.593 and 36.937. Apply Lloyd's 1892 rule for corrugated furnaces, which is,

$$\frac{1259\,(t-2)}{D} = \text{working pressure.}$$ Where

t = thickness in sixteenths of an inch.

(1) $$\frac{1259\,[(16\times .574)-2]}{35.593} = 254.1$$

(2) $$\frac{1259\,[(16\times .575)-2]}{36.937} = 245.4$$

and by dividing the actual collapsing pressures by these working pressures we get the factors of safety thus—

(1) $\frac{1400}{254.1} = 5.51$ (2) $\frac{1410}{245.4} = 5.74$

which gives a mean factor of safety of 5.63. But this is on a basis of steel having a tensile of 28.98 tons; consequently, the factor when reduced to a basis of 27 tons becomes

$$\frac{5.63\times 27}{28.98} = 5.24$$

All the remaining spots are calculated in exactly the same manner, and full particulars will be found in Table VIII.

I will now apply Mr. Milton's method of calculation to the factor of safety diagram, which represents the factor of safety on the furnaces *when new.*

Taking the symbols in the tables, we have

$$C = \frac{P\times D}{T} \text{ or } P = \frac{C\times T}{D}.$$

Also, let

p = Lloyd's working pressure $= \dfrac{\text{Lloyd's constant}\times(t-2)}{D}$

where,

t = thickness in sixteenths of an inch.

Hence, since $\frac{P}{p}$ = factor of safety,

$$\text{Factor of safety} = \frac{C \times T}{\text{Lloyd's constant} \times (t - 2)} \text{ or}$$

$$\text{Factor of safety} = \frac{C \times T}{\text{Lloyd's constant} \times (16\ T - 2)}$$

where,

T = thickness in inches.

If the factors of safety are now calculated by this formula, we shall get results identical with those given in Fig. 19 and as calculated above. For instance, referring again to Table II, we found that the factors of safety were—

(1) $\frac{1400}{254.1} = 5.51$ and (2) $\frac{1410}{245.4} = 5.74$

also, factor of safety $= \frac{C \times T}{\text{Lloyd's constant} \times (16\ T - 2)}$.

Substituting the values for C and T, we get—

(1) $\frac{86812 \times .574}{1259 \times [(16 \times .574) - 2]} = \frac{86812 \times .574}{1259 \times (9.184 - 2)} = 5.51$

(2) $\frac{90576 \times .575}{1259 \times [(16 \times .575) - 2]} = \frac{90576 \times .575}{1259\ (9.2 - 2)} = 5.74$

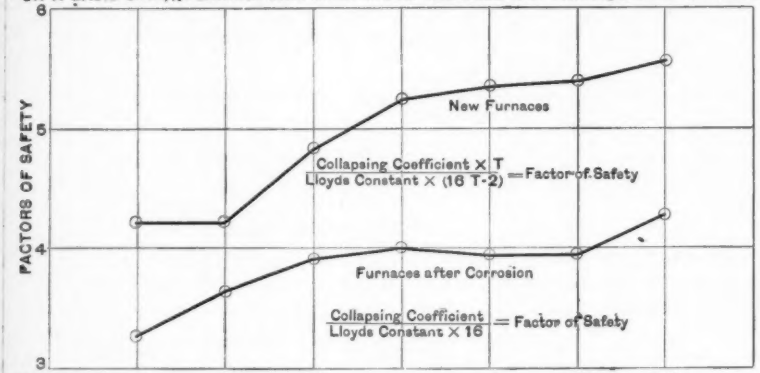
which are precisely the same results as above.

Mr. Milton explains that Lloyd's rule contains the expression $(t-2)$, where t is the thickness in sixteenths of an inch, and that this two-sixteenths represents an allowance for corrosion. Now, let us ascertain the factor of safety in the same furnaces when reduced one-eighth by corrosion. This makes the nine-sixteenth furnace into a seven-sixteenths, and reduces the diameter by $\frac{1}{4}$-inch. In furnaces of equal thickness we can take the nearest actual tests to the seven-sixteenths for the purpose, but in a Purves furnace there will be an undue advantage given to it by the additional material in the rib of a seven-sixteenths new furnace beyond that of a nine-sixteenths when corroded two-sixteenths. As it is necessary to deal with the actual test results, however, we may ignore this advantage thus gained by the Purves, and then calculate the factors of safety as follows:

The equation becomes—

DIAGRAM A

DIAGRAM OF FACTORS OF SAFETY CALCULATED BY LLOYDS FORMULAE ON A BASIS OF A 5/16 FURNACE AND ALSO WHEN REDUCED 1/16 IN THICKNESS BY CORROSION.

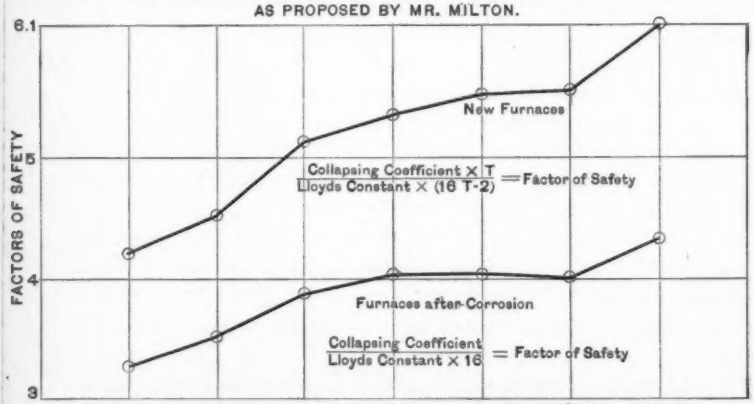

6
5
4
3
FACTORS OF SAFETY
New Furnaces
Collapsing Coefficient × T / Lloyds Constant × (16 T-2) = Factor of Safety
Furnaces after Corrosion
Collapsing Coefficient / Lloyds Constant × 16 = Factor of Safety

RIGINAL THICKNESS=	.557	.612	.581	.5745	.536	.5935	.5485
HICKNESS AFTER CORROSION	NO TESTS	.534	.4635	.460	.452	.444	.461
	PURVES 1886	PURVES 1887	PURVES 1889	FOX 1892	HOLMES 1891	EARNLEY 1888	MORISON 1892

FACTORS CALCULATED ON STEEL HAVING A TENSILE STRENGTH OF 27 TONS PER SQUARE INCH.

DIAGRAM B

DIAGRAM CALCULATED ON THE MEANS OF ALL THE FURNACES EXCEPT THE 5/16 PURVES AS PROPOSED BY MR. MILTON.


6.1
5
4
3
FACTORS OF SAFETY
New Furnaces
Collapsing Coefficient × T / Lloyds Constant × (16 T-2) = Factor of Safety
Furnaces after Corrosion
Collapsing Coefficient / Lloyds Constant × 16 = Factor of Safety

TWO TESTS PURVES 1886	FOUR TESTS PURVES 1887	FOUR TESTS PURVES 1889	THREE TESTS HOLMES 1891	SIX TESTS FOX 1892	FIVE TESTS EARNLEY 1888	SIX TESTS MORISON 1892

FACTORS CALCULATED ON STEEL HAVING A COMMON TENSILE STRENGTH OF 27.2 TONS PER SQ. INCH.

Bradley & Porter, Engr's, N.Y.

$$\text{Factor of safety} = \frac{C \times T}{\text{Lloyd's constant} \times (t - o)}$$

$$= \frac{C \times T}{\text{Lloyd's constant} \times (16\frac{1}{2}T - \frac{1}{2}O)} = \frac{\text{Collapsing co-efficient}}{16 \times \text{Lloyd's constant}}$$

which is exactly the equation Mr. Milton uses.

In order that there may be no assumptions, I will take the nearest actual tests to seven-sixteenths, and we will refer again to Table II. The tests nearest seven-sixteenths in thickness are the 3d and 4th, viz., .452 and .468. The collapsng co-effiicient will be—

(1) $\frac{35.625 \times 1130}{.452} = 89000$ (2) $\frac{36.469 \times 1090}{.468} = 85000$

and reduced to steel of 27 tons—

(1) $\frac{89000 \times 27}{29.26} = 82200$ (2) $\frac{85000 \times 27}{29.02} = 79000$

$$\text{Factor of safety} = \frac{C}{16 \text{ Lloyd's constant}} = \frac{C}{16 \times 1259}$$

(1) Factor $= \frac{82200}{16 \times 1259} = 4.08$ (2) Factor $= \frac{79000}{16 \times 1259} = 3.92$

which gives a mean of 4.0. All the other tests have been treated in the same way, and the results are shown in Diagram *A*.

In addition to Diagram *A*, I have calculated a second diagram *B* on the *means* of all the furnace tests except the five-sixteenths Purves, as proposed by Mr. Milton, and I have reduced all these to a basis of steel having a tensile strength of 27.2 tons per square inch, which is Mr. Milton's basis.

Taking the Fox tests with Lloyd's 1892 rule, as before, we have for new furnaces—

$$\frac{C \times T}{1259\,(16\,T - 2)} = \text{Factor of safety,}$$

and calculating the factors of safety by this formula we get the following: 6.65, 5.68, 6.12, 5.75, 5.5, and 5.74, which give a mean of 5.9; and, reducing this to steel of 27.2 tons tensile, we get—

$$\frac{5.9 \times 27.2}{29.07} = 5.52$$

For furnaces after corrosion has taken place to the extent of one-eighth, the formula becomes—

$$\frac{C}{1259 \times 16} = \text{Factor of safety.}$$

The mean collapsing co-efficient for the six furnaces is 85691, and, substituting this value, we get—

$$\frac{85691}{1259 \times 16} = 4.25$$

and reduced to steel of 27.2 tons tensile—

$$\frac{4.25 \times 27.2}{29.07} = 3.98$$

All the other spots have been calculated in exactly the same manner.

I may add that all my calculations have been made with the slide rule, so that possibly, in some cases, the second and third decimal places in the factors of safety may not be quite accurate. However, this would not materially affect the diagrams.

In comparing the formulæ of the Board of Trade and Lloyd's, it will be seen that the $(t—2)$ in the rule of the latter has the effect of giving a premium to thick furnaces. For example, take a Purves furnace 36 inches inside diameter and seven-sixteenths thick; the working pressure allowed by the Board of Trade for steel from 26 to 30 tons would be 166.3, and by Lloyd's 157.5; or a difference of 8.8 pounds per square inch in favor of the Board of Trade.

Now, take the same furnace five-eighths thick. The Board of Trade working pressure would be 235 and Lloyd's 249, being a difference of 14 pounds in favor of Lloyd's, or a total difference of 22.8 resulting from the application of Lloyd's formula. But in the Purves design, we know that the thicker furnace is relatively the weaker, so the wisdom of the rule applied to this particular design is doubtful. Mr. Milton explains this $(t—2)$ in its general application to all furnaces by saying that the strength of a thick furnace is affected less by corrosion than a thin one, but when it is remembered that modern furnaces have a thickness ranging only from about $\frac{1}{2}$ inch to $\frac{3}{4}$ inch, it would seem preferable that the factor of safety should be calculated on the furnaces when new and made sufficiently large to allow for corrosion.

In concluding my reply to Mr. Milton, I wish particularly to

state that I do not in any way criticise the amount of the factors of safety, as there is no doubt whatever that the factor of safety on furnaces is less than in any other part of the boiler. In 1888, when the Engineer-in-Chief of the Board of Trade reported on the successful working of 27,000 Fox's furnaces made between 1879 and 1888, and manufactured of low tensile steel, we had had practically 80 pounds steam pressure and little or no forced draft. Now, we have pressures of 160 pounds to 200 pounds and a good deal of forced draft; consequently the factor of safety should be even greater now than then.

Take for example the American Liners. It is quite a common occurrence for some of the furnaces to be set up at the end of a voyage, showing the factor of safety has entirely disappeared; consequently, I am certainly of opinion that in forced draft boilers especially, where the temperature is excessive and the liability to scale correspondingly great, the factors of safety allowed both by the Board of Trade and Lloyd's are by no means too large. My contention is not that they are too large in amount, but that they are not equal for all designs of furnaces, and that they should be made equal in justice both to users and manufacturers. Another point is, it would appear to be desirable that the factors of safety should be based on the lowest limit of the range of tensiles; for example, take the ranges to be from 23 to 26 tons and from 26 to 30 tons, the factor of safety should undoubtedly be based on the 23 tons in the former case and on the 26 tons in the latter, and not on the means of the range.

THE TRANSMISSION OF HEAT THROUGH TUBE PLATES.

By A. J. Durston, Engineer-in-Chief of the British Navy.

[Paper read before the Institution of Naval Architects, March 23, 1893.]

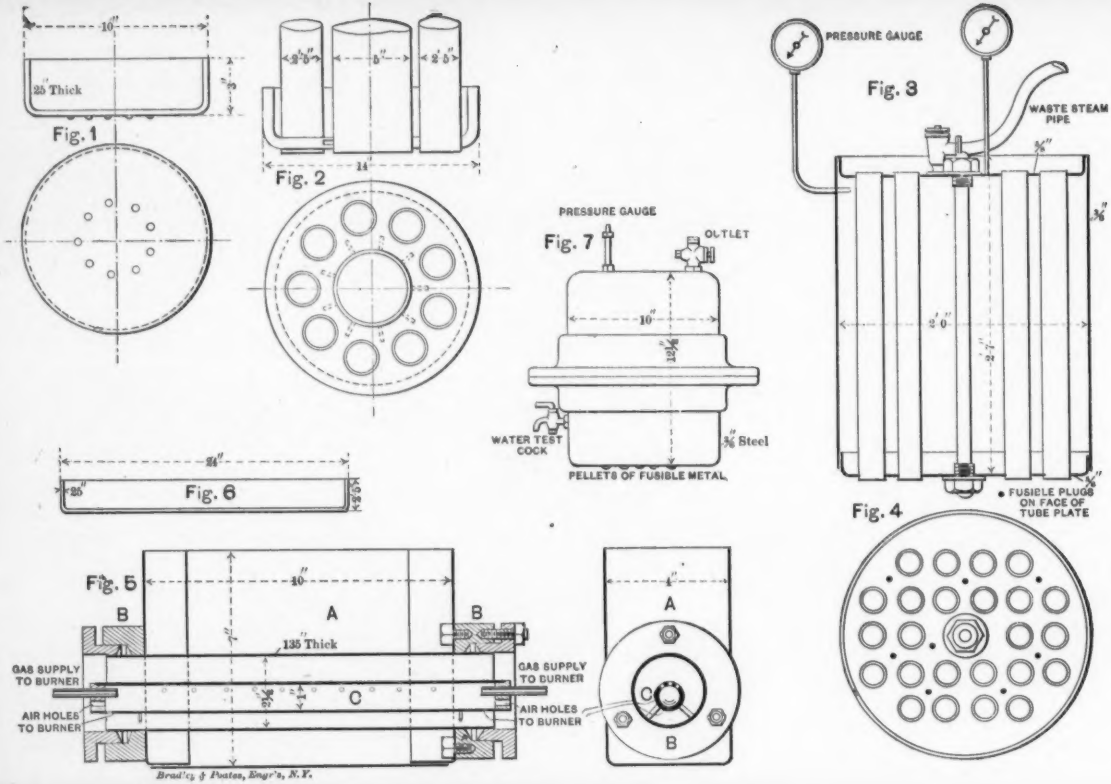
During the past three years, various experiments have been made at Devonport with the view of ascertaining the temperature of tube plates and tubes under certain conditions of working, and their bearing on the leakage of tubes, the results of which, it is hoped, may be of interest to the members of this Institution.

Although these experiments are not complete and will be carried further, it is submitted the record of what has been obtained up to the present will, in addition to being of interest to members, secure their valuable co-operation in making similar experiments, such as those made by the late Dr. Kirk, and published in "Engineering" of July 15, 1892, and thus enlarge the sources of our information and tend to throw more light on the important subject of the limit of working endurance of tube and plate-heating surfaces, and of the materials of which they are made.

The first experiments were made in 1890, as follows:

1. *To ascertain the temperature of the hot side of a plate through which heat is passing to boiling water.*

For this purpose a circular flanged dish, 10 inches in diameter and 3 inches deep (see Fig. 1), had attached to the bottom eight pieces of fusible solder of different compositions, the melting points ranging from 220 degrees Fahr. to 250 degrees Fahr. The vessel was half filled with water, and placed over a Bunsen gas flame, the temperature of which was about 1,500 degrees Fahr., and allowed to remain until the water had been boiling freely some time. It was then found that the alloys whose fusing points ran up to 240 degrees Fahr. melted, but the next, which


10"
.25" Thick
Fig. 1
Fig. 2
14"
PRESSURE GAUGE
Fig. 3
WASTE STEAM PIPE
PRESSURE GAUGE
Fig. 7
OUTLET
10"
WATER TEST COCK
⅜" Steel
PELLETS OF FUSIBLE METAL
24"
25"
Fig. 6
Fig. 4
FUSIBLE PLUGS ON FACE OF TUBE PLATE
Fig. 5
10"
B
A
B
.135" Thick
GAS SUPPLY TO BURNER
GAS SUPPLY TO BURNER
C
AIR HOLES TO BURNER
AIR HOLES TO BURNER
4"
A
B
Brad'ley & Poates, Engr's, N.Y.

would fuse at 243 degrees Fahr., only slightly softened. This temperature of that side of the plate was therefore assumed to be about 240 degrees Fahr.

A layer of grease obtained from the interior of the boilers of a new ship was next spread about $\frac{1}{32}$ inch thick over the inside of the bottom of the vessel and the previous experiment repeated. The temperature of the outer surface of the plate was this time shown by fusible alloys to be about 330 degrees Fahr., or a rise of 90 degrees Fahr., due to the presence of the layer of grease. The increase of temperature was not so large as was expected, and the experiment on being repeated gave virtually the same results.

2. To ascertain the temperature at the center of its thickness of a plate resembling a boiler tubeplate exposed to a forced blast fire.

A flanged $\frac{3}{4}$-inch plate was fitted with short lengths of steel boiler tube, as shown in the sketch, Fig. 2, the center tube being of large size to allow of drilling $\frac{3}{16}$-inch holes in the center of the thickness of the plate. In these holes were placed square pieces of fusible alloys, and the tubes rolled as usual. Water was then put in nearly to the depth of the flange, and the apparatus placed over a forge fire, the blast being used, and the temperature of the fire being about 2,000 degrees Fahr. The experiment was continued for about half an hour, fresh water being supplied to replace that boiled away. It was found that the alloys whose fusing points ran up to 290 degrees Fahr. had melted, but the next, which would melt at 336 degrees Fahr., was unchanged. The temperature of the plate at the center was therefore taken to be between 290 degrees Fahr. and 336 degrees Fahr., a greater temperature even than on the fire side of the plate in the previous experiment when using the Bunsen flame (1,500 degrees).

3. To determine the temperature of the tubeplate at which injurious overheating (i. e., such as to cause leaky tubes) takes place.

For this purpose a small boiler was made, as shown in Figs. 3 and 4, page 395, in which there were twenty-four tubes, $2\frac{1}{2}$ inches in diameter, the tubeplate being $\frac{3}{8}$ inch thick. Of these tubes eight were brass, seven steel, and nine iron, so that any

difference in the behavior of the different materials might be observed, the tubes being as far as possible grouped in threes (one of each), in order to insure their being exposed to the same conditions. A calculated amount of water was put into this boiler, such that, when wholly evaporated, the internal pressure would be 100 pounds on the square inch, means being fitted to prevent this pressure being exceeded by the expansion of the steam gas. The boiler was placed over a forge with the tubes vertical, and the blast put on and continued until the tubeplate showed a red heat, equal to about 1,400 degrees Fahr. The pressure of steam inside was at first 100 pounds, but it fell as the plate overheated. The boiler was then removed from the fire and allowed to cool.

On testing with water, all the tubes leaked so badly that no pressure could be obtained. No difference was noticed in the behavior of the brass, steel or iron tubes. This indicated that raising a tubeplate to red heat causes tubes of all these materials to leak very badly.

Next a number of holes were drilled and tapped partly through the plate, and plugs of lead inserted. The previous experiment was repeated, the boiler being removed from the fire as soon as the lead melted, it being assumed that the plate was then at about the same temperature as melting lead—viz: 630 degrees Fahrenheit. The boiler was afterwards tested by water pressure to 200 pounds on the square inch, practically without any leaks. The tapped holes were next filled with zinc plugs, and the experiment repeated, the steam pressure being 80 pounds on the square inch. Two of the brass tubes split during this experiment, and, on testing the boiler after replacing them by new, it was found necessary to roll two other brass tubes, after which the boiler was practically tight up to the test pressure of 200 pounds, the iron tubes being without a weep, and a few of the brass and steel tubes leaking very slightly. It is, therefore, assumed that a tubeplate, to be overheated sufficiently to make the tube-joints leak to an appreciable extent, must be raised at least to the temperature of melting zinc—viz: 750 degrees Fahrenheit.

4. *To ascertain the loss of efficiency of the heating surface of tubes in a boiler, due to a thin coating of grease deposit.*

A tube taken from a boiler of a new ship was cut into lengths, and tried in the apparatus show in Fig. 5, annexed. This consisted of a rectangular iron vessel *A* to hold the water, with holes and stuffing-boxes *B*, for the piece of boiler tube under test, the heat being supplied by a horizontal Bunsen burner *C*. The results of the experiments showed that the thin coating of grease deposited on these tubes during the ship's trials caused a loss of efficiency as heating surface, as compared with a perfectly clean tube, of from 8 to 15 per cent., the mean of many experiments giving 11 per cent.

During the past year (1892) some of these experiments have been repeated on an extended scale as follows:

5. *Temperature of plates when boiling water in an open vessel under various conditions corresponding to previous experiment* (1).

A larger vessel than before, being a flanged dish 2 feet in diameter, 2½ inches deep, and ¼-inch thick, was used, and a constant supply of water maintained. The vessel was placed over a forge fire instead of over a Bunsen burner. With a moderate blast the temperature of the hot side of the plate was as before found to be 240 degrees Fahrenheit, when boiling fresh water. More blast was then applied, and the temperature of the plate went up to 280 degrees Fahrenheit. This experiment was extended by repeating it with various foreign substances in the water, with the results shown in the table:

	Temperature of hot side of plate. Deg. Fahr.	Temperature of fire. Deg. Fahr.
Clean fresh water	280	2200
* Mineral oil gradually added up to 5 per cent	310	2300
Fresh water, with 2½ per cent. of paraffin	330	2100
Fresh water with 2½ per cent. of methylated spirits	300	2500
† A greasy deposit $\frac{1}{16}$ in. thick on plate	above 550	2500

* Genuine American distilled oil, with the paraffin scale extracted, so that the oil will remain fluid at 32 deg. Fahr., with a minimum density of .91 and a minimum flash point of 400 deg. Fahr.

† Other experiments showed that this temperature varies greatly, and depends chiefly on the nature and thickness of the deposit.

6. *Temperature of plates when boiling water under various conditions at a higher temperature than* 212 *deg. Fahr.*

The higher temperature was obtained by using a closed vessel, as shown in Fig. 7, annexed, and boiling the water at pressures above atmospheric. The following results were obtained:

a. Using clean water and surfaces:

	Temperature of hot side of plate. Deg. Fahr.	Temperature of water. Deg. Fahr.	Difference. Deg. Fahr.
Over Bunsen burner	430	363	67
Over blast forge (full blast)	430	344·5	85·5

b. Bottom of vessel coated with grease:

	Temperature of hot side of plate. Deg. Fahr.	Temperature of water. Deg. Fahr.	Difference. Deg. Fahr.
Over forge fire—grease deposit $\frac{1}{16}$ in. thick.	510	359	151
Ditto, but using grease of drier or earthier nature	550	351	199
* Ditto, and spreading the grease up the sides of vessel as well	617†	80	537

*The grease was spread 1½ in. up the sides of the vessel from the bottom, being then 1 in. below the water level.

†This temperature was reached three minutes after placing the vessel on the fire with the blast turned on.

7. *Experiments showing the behavior of tubes of various materials.*

The last-mentioned experiments were repeated several times with the apparatus shown in Fig. 3, annexed, except that the tube-plate was $\frac{5}{8}$ inch thick, and tubes of copper, brass, iron and mild steel were tested simultaneously. It was not thought necessary, however, to insert the particulars of these experiments, which varied considerably, as they were repeated on a larger and more reliable scale, as described in Experiments 9 and 10, but it is worthy of remark that upon all occasions some brass and copper tubes leaked even when the plate was below the temperature of melting lead, viz., 617 degrees Fahr., showing that these materials did not stand so well as iron or steel, whilst between the latter materials—Staffordshire as well as Lowmoor iron tubes being used—there appeared to be practically no difference.

HEATING SURFACE, TUBES_____219 SQ. FT.
" " TUBE PLATES_10.5 " "
" " TOTAL_____229.5 " "
GRATE AREA_____6.25 " "
WEIGHT OF WATER CONTAINED IN BOILER UP TO WORKING LEVEL } 2460 LBS.

Fig. 8

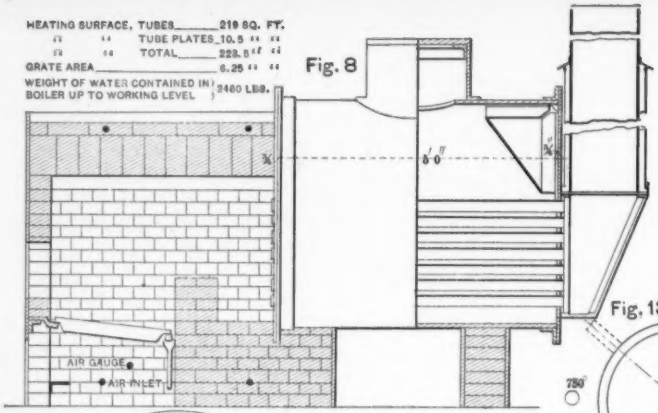


Fig. 9

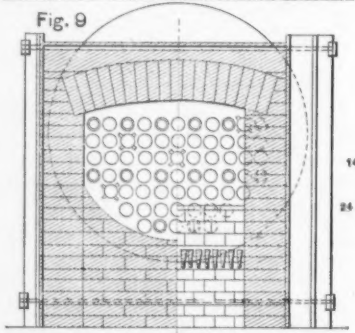


Fig. 11

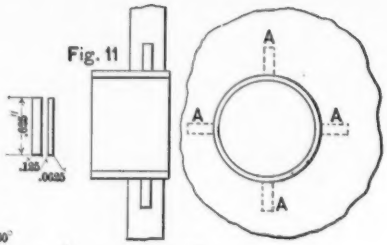


Fig. 13

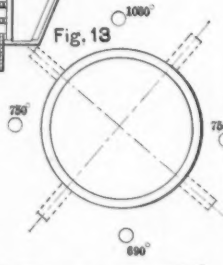
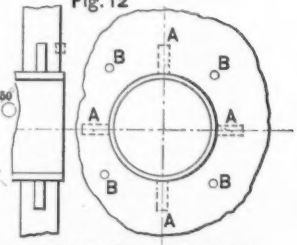


Fig. 12



A. FUSIBLE PELLETS IN MIDDLE OF PLATE.
B. FUSIBLE PELLETS IN FACE OF PLATE.

Fig. 15

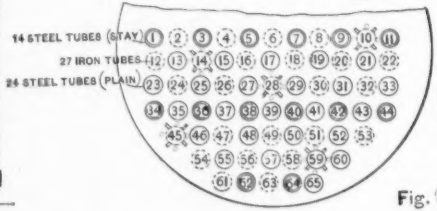
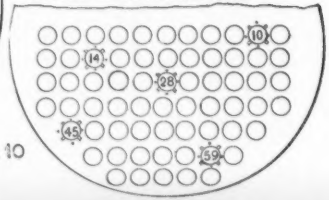


Fig. 10



8. *Experiments to determine whether at higher pressures of steam there is any marked addition to the excess of temperature of the hot side of the plate over that of the boiling water.*

These were carried out with the closed vessel shown in Fig. 7. A large number of results were obtained, the details of which showed that there is no marked addition to the excess of temperatures at the higher pressures.

9. *Experiments on the temperature of the center of the thickness of a tubeplate, with an experimental boiler working with closed ashpits and moderate air pressure.*

	Mean.	*Maximum.*
Steam pressure	143	150
* Air pressure in closed ashpit, in inches of water	.3	.5
† Temperature of combustion chamber, degrees F	2,850	3,100
Temperature in tubes (middle of length), degrees F	1,550	1,800
Temperature in smoke-box, degrees F	1,400	1,600
Coal used per hour, pounds	188	
Coal used per square foot of grate, pounds	30	
Water evaporated per hour, pounds	1,039	
Water evaporated per hour per square foot of tube and tube-plate surface, pounds	4.62	

* The highest air pressure with the fan then available.
† Temperatures measured by a Le Chatelier Pyrometer.

Figs. 8 and 9, annexed, show this experimental boiler with its built-up brick furnace. The tubeplates are bolted to the shell, so as to admit of being easily removed and replaced by others. As first fitted, the draft was supplied through a closed ashpit, and under these conditions a two hours' trial was made, of which particulars are given in the table. Five of the tube holes (see Fig. 10)—Nos. 10, 14, 28, 45 and 59—had each four pieces of fusible alloy $\frac{5}{8}$ inch long let into the plate at the middle of its thickness, as shown in Fig. 11. The condition of these at the conclusion of the trial is given on the next page; the maximum temperature obtained in the combustion chamber was 3,100 degrees Fahr., and the maximum temperature of the steam 366 degrees Fahr.

From the above it will be seen that, of the pieces of fusible alloys placed in the tubeplate, all of the six whose melting points

run from 435 deg. up to 490 deg. Fahr. had fused completely. The ten ranging from 540 deg. upwards remained unchanged. Of the intermediate ones, those having melting points of 500 deg. and 510 deg. remained unchanged, whilst two others with melting points of 520 deg. and 530 deg. were fused just at the ends near the tubes. It would appear, therefore, that the temperature of the plate at the middle of its thickness did not rise to 540 deg. Fahr., but at some of the tube joints it rose to 530 deg. Fahr.

Number of tube.	Melting point of alloy, deg. Fahr.	Condition after trial.
59	435	Fused completely.
	450	Fused completely.
	460	Fused completely.
	470	Fused completely.
45	480	Fused completely.
	490	Fused completely.
	500	Not fused.
	540	Not fused.
10	510	Not fused.
	520	Fused at end next tube.
	530	Fused at end next tube.
	540	Not fused.
28	550	Not fused.
	617	Not fused.
	680	Not fused.
	773	Not fused.
14	550	Not fused.
	617	Not fused.
	680	Not fused.
	773	Not fused.

10. *Further experiments to ascertain the temperature of fire side and middle of thickness of tubeplate in experimental boiler with forced draft and closed stokeholds.*

Due to the difficulty in stoking the boiler arranged as previously described, the flames blowing out into the stokehold,

unless the draft was shut off the ashpit each time of firing, the boiler was inclosed in an air-tight stokehold, and the draft supplied by a fan and engine of increased dimensions, in order to obtain a higher rate of combustion. Under these altered conditions, the trials were continued, and, in addition to the fusible alloys let into the middle of the plate, four pieces, $\frac{5}{32}$ in. in length and $\frac{3}{16}$ in. in diameter, were fitted into the face of the plate, around each of the tubes mentioned and as shown in Fig. 12. These pellets projected $\frac{1}{32}$ in. beyond the plate.

The following table shows the observations made in four trials with this boiler:

Table A.

DATA OF TRIALS.

Number of trial.	First.	Second.	Third.	Fourth.
Duration of trial.....hrs...	5	5	5	3.8
Pressure of steam.....lb...	145	142	140	144
Air pressure in stokehold.....in...	3	3 for first two hours; 3½ for next three hours.	3	2.9
Total coal used during trial.....lb...	2,800	3,188	2,632	Not accurately taken.
Total water evaporated.....lb...	14,125	14,775	13,148	10,276
Coal per square foot of grate per hour .....lb...	90	102	84.2	
Water evaporated per square foot of tube and tube-plate surface per hour.....lb...	12 64	13.22	11.76	11.99
Temperature in combustion chamber.....deg. Fahr...	2,750	2,500	3,100	3,200
Amount of mineral oil used.....lb...	.....	.....	9	5*
Oil used in percentage of feed.....	.....	.....	.07	.05

* Boiler not cleaned out from previous trial.

On the first trial (five hours with 3-inch air pressure, using clean feed water) sixteen of the pellets in the face of the plate were made with melting points from 490 degrees to 690 degrees Fahr., and the remaining four were of antimony (melting point 1,060 degrees Fahr.) All were melted except the four antimony.

On the second trial (five hours with 3-inch air pressure during the first two hours, and 3½-inch during the next three, using

clean feed water) the pellets placed in the face of the plate around each of the five tubes were as shown in Fig. 13, viz., one of antimony (1,060 degrees), two of zinc (750 degrees), and one of alloy melting at 690 degrees. Of these the five antimony and three of zinc (at tubes 14, 45 and 59) remained intact, all the rest melted.

On the third trial, which was of five hours' duration with 3-inch air pressure, with the pellets arranged as on the preceding trial, a total quantity of 9 pounds of oil was admitted to the boiler mixed with the feed. The five antimony and one (at tube 45) out of the ten zinc plugs remained intact, all the rest melted. As this was the last trial at which the tubes remained tight, the temperature of the plate should be noted for comparison with the next trial when the tubes gave out. Since all the zinc pellets melted (except one low down at the edge of the tubeplate), the temperature of the tubeplate must have been above that of melting zinc (750 degrees) and below that of antimony (1,060 degrees). On the fourth trial, which was a continuation of the third, an additional five pounds of oil was admitted with the feed water. During this trial the tubes gave out when cleaning fires after $3\frac{5}{6}$ hours' run. The pellets in the tubeplate were arranged as before, and melted as follows:

Around tubes 10, 14 and 28 (hottest part of the plate) all the zinc and alloys melted; the antimony partly melted in Nos. 14 and 28, but remained intact in No. 10. Around tubes 45 and 49 the antimony and zinc remained intact. This shows that the plate was about the temperature of 1,060 degrees Fahr., at all events during the latter part of this trial, and it is presumed that the tubes then gave out; whereas, from the results of the third trial, the tubes remained quite tight up to and above the temperature of melting zinc, 750 degrees Fahr.

The five tubes were now drawn, and the fusible alloys which had been let into the plate at the center of its thickness were examined. They were made with melting points ranging from 480 degrees to that of zinc, 750 degrees Fahr., and arranged as shown in Fig. 14. All were found to be melted, except the zinc at tubes 14 and 28.

Comparing the highest temperature shown at the face and center of the plate around the tubes most exposed to heat, the temperatures are as follows:

Tubes Nos. 14 *and* 28.

Temperature at face of plate:

Antimony partly melted, *i. e.*, 1,060 degrees Fahr.

Temperature at middle of plate:

680 degrees alloy melted; zinc not melted; *i. e.*, between 680 degres and 750 degrees Fahr.

11. *Experiments as to behavior of Lowmoor iron versus steel tubes as regards leakage in experimental boiler.*

The series of trials above described with the boiler shown in Figs. 8 and 9, were utilized for this purpose, the tubes being arranged as shown in Fig. 15, so that, as far as practicable, Lowmoor iron and steel tubes were placed alternately. Particulars of the four trials made are those given on Table A.

First trial of five hours' duration with 3-inch air pressure. No leakage of tubes occurred at this trial.

Second trial of five hours' duration with 3-inch air pressure for first two hours, and $3\frac{1}{2}$ inches for next three hours. At the conclusion of this trial the fan was kept going for some time after drawing the fire, but no leakage of tubes occurred. Attention is called to this fact, as great stress is frequently laid on the action of cold currents of air in producing leaky tubes.

Third trial of five hours' duration with 3-inch air pressure, a certain proportion of mineral oil being mixed with the feed water. On this trial no leakage of the tubes occurred.

The fourth trial was proposed to be an eight hours' continuation of the last, using oil in the proportion shown on the table. At this trial leakage of tubes occurred under the following circumstances:

Just before the end of the fourth hour the fire was being burned down to clean it, in order to be able to work for eight hours, the air pressure being gradually reduced. Ten minutes after beginning to burn down, and with the air pressure at 1 inch, the tubes gave out, and began to leak badly. On examination of the boiler, it was seen that six Lowmoor iron tubes and one steel tube

leaked badly; three Lowmoor tubes and nine steel tubes leaked slightly. On applying water pressure it was found that at 60 pounds all the tubes in the boiler leaked, more or less, except No. 34 and the tubes in the two bottom rows, which were probably protected by ashes, &c., during the trial; but the three Lowmoor tubes, Nos. 14, 16 and 18, were leaking most.

The results of these trials, therefore, appear to show that Lowmoor iron tubes are, at least, not superior to steel ones; and, as from experience we know the latter will stand more rolling than the ordinary iron tube of commerce, it justifies our preference for steel tubes.

12. *Results obtained with grease in boilers.*

As bearing on the important point of the presence of grease in boilers, it may here be stated that in one of the yard boilers at Portsmouth the furnace crown came down shortly after concluding some experiments in using greasy water. In a similar boiler at Devenport, the Adamson joint of the furnace gave out after similar treatment. These boilers were ordinarily worked at about 60 pounds pressure, and used only fresh water from the yard mains, and had experienced no serious defect till the introduction of grease experimentally. This is again borne out by repeated examples in Navy boilers of all types of the deleterious effect of the presence of grease in them. Referring back to the former part of this paper, the five experiments on page 439 and the three experiments on page 440 also show how very great is the effect of grease on the water side of the plate-heating surface. These are also confirmed by the trials with the experimental boiler, which, although subjected to the high temperature of about 3,000 degrees Fahr. in the combustion chamber and 1,600 degrees Fahr. in the smoke-box, and further subjected to hard treatment by admission of cold air through the tubes after drawing the fire at the conclusion of the second trial, did not leak till grease was used in it.

13. *Experiments on the temperature at various parts of the tubes of an ordinary marine boiler.*

The following experiments were made to ascertain the fall of temperature of the products of combustion in passing through

Fig. 18

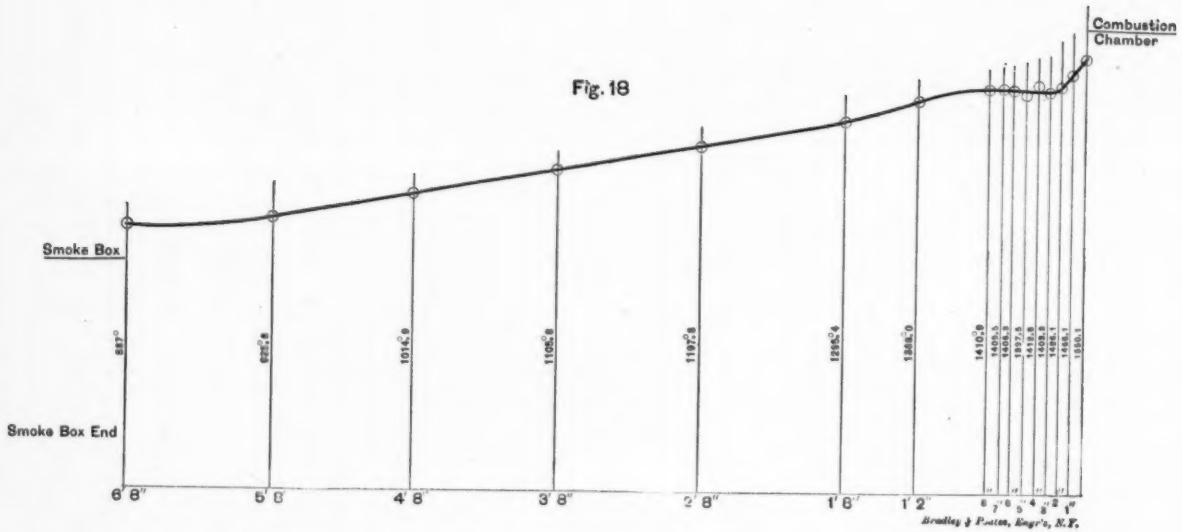

Combustion Chamber
Smoke Box
Smoke Box End
852°
822.8
1014.9
1105.8
1197.8
1295.4
1365.0
1410.9
1409.5
1408.8
1407.5
1410.8
1408.8
1408.1
1408.1
1359.1
6' 8"
5' 8"
4' 8"
3' 8"
2' 8"
1' 6"
1' 2"
6
7' 6
5
4
3
2
1"
Bradley & Paxton, Engr's, N.Y.

Fig. 14.

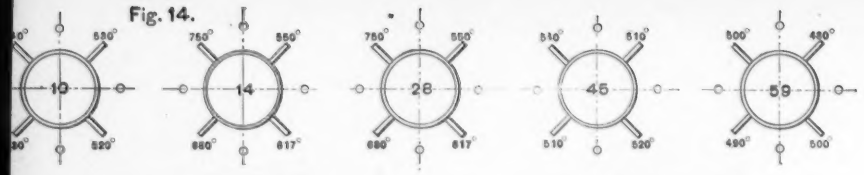


Fig. 16.

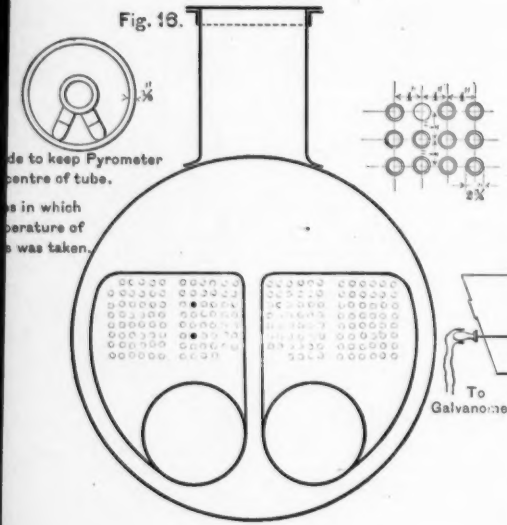
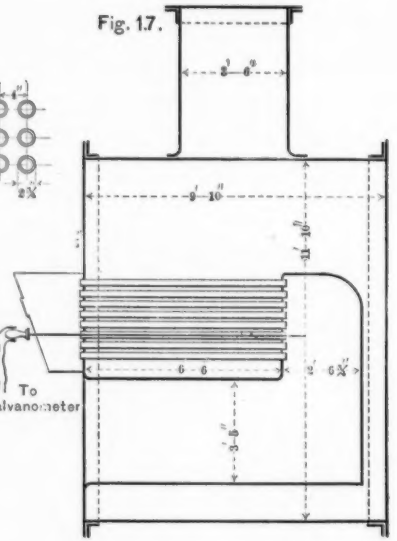
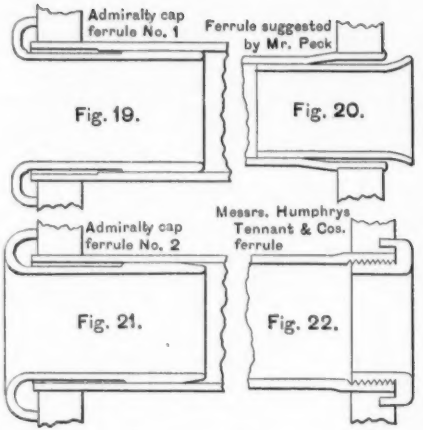


Fig. 17.





the tubes of an ordinary marine boiler used in Keyham Yard. This boiler (see Figs. 16 and 17) had two furnaces and 166 return tubes, $2\frac{3}{4}$ inches in external diameter, and 6 feet 8 inches long from outside to outside of tubeplates. The temperatures were taken by a Le Chatelier thermo-electric pyrometer, the junction of which was passed through the smokebox and down the center of the tubes, as shown in the sectional elevation. Tubes in a vertical row nearly over the center of one of the furnaces were taken for the experiment. The temperature was recorded at each foot length until near the combustion chamber, where it was taken at every inch. The boiler was being worked at its normal capacity, the consumption of coal being about 17 pounds per square foot of grate.

The following are the mean results of eight sets of records:

	Degrees Fahr.
Temperature in combustion chamber,	1644
" just inside tube,	1550
" in tube 1 inch from combustion chamber,	1466
" " 2 inches, "	1426
" " 3 inches, "	1405
" " 4 inches, "	1412
" " 5 inches, "	1398
" " 6 inches, "	1406
" " 7 inches, "	1400
" " 8 inches, "	1410
" " 1 foot 2 inches, "	1368
" " 1 foot 8 inches, "	1295
" " 2 feet 8 inches, "	1198
" " 3 feet 8 inches, "	1106
" " 4 feet 8 inches, "	1015
" " 5 feet 8 inches, "	926
" " 6 feet 8 inches, "	887
" in smokebox,	782

These results are plotted out and shown graphically on Fig. 18 annexed. It will be seen that about two-elevenths of the total fall of temperature takes place in the first two inches, after

which the fall is fairly evenly distributed throughout the length of the tube. The above shows that even beyond 6 feet in length there is an appreciable transfer of heat, but a further experiment will be made by shortening these tubes by 1 foot and lengthening the combustion chambers by the same amount, and making comparative trials as compared with the boiler as now arranged.

The above concludes, so far as we have gone, what may be termed the experiments made on a small scale, and an account will now be given of some experiments on a larger scale on shipboard, with the object of avoiding leakage of boiler tubes.

It is a matter of common knowledge that the Admiralty has experienced considerable trouble with leakage of tubes in the double-ended common combustion chamber boilers and those of the locomotive type.

Of the former there are three kinds, distinguished by having either two, three or four furnaces at each end.

As regards the two-furnace type, various expedients have been tried in one or other of the ships so fitted to overcome the leakage. These may be summarized as follows: Rolling tubes with a shoulder inside the tubeplate. Beading tubes over the tubeplate. Rolling the tubes parallel. Fitting ordinary ferrules in tubes. Shortening the grates, involving an increase in the air pressure. Replacing the stays from the top of the combustion chamber to the shell of the boiler by dog stays having no connection with the top of the boiler. No definitely beneficial results have attended any of these measures.

The modification in these boilers that gave the greatest benefit was that of removing two vertical rows of tubes over the center of each furnace, as suggested to me by Mr. Seaton, of Messrs. Earle's, and this has been applied to all the boilers of that type. It has also been applied to those having three and four furnaces at each end, but the results in these cases were not nearly so satisfactory. In these boilers, the expedients of shortening the grates and easing the stays immediately above and below the tubes, so as to allow a slight movement of the tubeplates with the expansion of the tubes, have also been tried, but with no

permanent benefit. Continued experience with the *Thunderer* and *Vulcan*—vessels having this double-ended common combustion chamber boiler with three furnaces at each end—was so unsatisfactory, both on trials and on actual service afloat in the former ship, that the reboilering of these ships, as well as the *Devastation*, was provided for in the navy estimates for the current financial year. In all these boilers, the combustion chamber was divided by brick walls into a separate combustion chamber to each furnace, or to each two adjacent furnaces in the eight-furnace boilers. In the *Thunderer* and *Devastation*, from the arrangement of the nests of tubes, it was only possible to divide the combustion chamber into two parts, one to each end of the boiler.

Concurrently with the efforts that were being made to overcome the leaky tube troubles by improving the circulation, experimental trials were being conducted in the locomotive boilers of the torpedo gunboat class: (1) By plastering the tubeplate with a non-conducting composition, thus protecting it on the fire side, and (2) by ferruling the tubes with fireclay cap ferrules, the caps of which afforded protection to the tube ends and the larger part of the tubeplate.

The object of these experiments was not to demonstrate the effectiveness of the materials used, but to show whether the leakages of boiler tubes were not due to the overheating of the tubeplate and ends. This was established, for as long as, in the first case, the cement adhered, and, in the second case, the ferrules lasted, leakages did not occur. They were, however, both liable to rapid destruction, and could not be relied on as a permanent protection. Numerous devices having a similar object were, from time to time, suggested, but all had practical difficulties preventing their adoption.

Amongst the first of the practical suggestions made for ferruling the tube ends was that patented by Messrs. Humphrys, Tennant & Co., and illustrated by Fig. 22. It will be seen that this ferrule is screwed into the tube at the firebox end, and that the cap fits into an annular recess cut in the tubeplate. The principle of this ferrule is that, when a contraction in diameter takes

place, due to variations of temperature, the outer part of the ferrule tends to tighten upon the concentric portion of the tube-plate. Further, as the ferrule is screwed into the tube, it has the advantage of the holding power afforded by the rolling of the tube into the cooler smokebox tubeplate. It will also be seen that from its construction it provides a large amount of jointing surface, and an intricate passage to prevent the escape of water. On the other hand, it has the disadvantage that tubes cannot be withdrawn for cleaning and repairs, but must be cut out, and it is somewhat costly in fitting. Messrs. Humphrys' proposal to fit these ferrules in the two after six-furnace, common-combustion-chamber boilers of the *Medea* was approved.

For these trials, iron tubes were fitted in the starboard boiler, and steel tubes to the port boiler. The modifications that had already been made in the furnaces, combustion chambers, and tubes were allowed to stand for these trials. They consisted in shortening the grates of the middle furnaces by the removal of the back tier of bars, dividing the common combustion chamber by brick walls to form a separate combustion chamber to each furnace, and the removal of vertical rows of tubes over the furnaces.

An eight hours' trial of both boilers was made at natural-draft power, which was satisfactorily obtained with a mean air pressure of 1.26 inches. The examination of the boilers showed that about 70 tubes in all had leaked very slightly. The ferrules had practically not scaled by oxidation. The leaks were so trivial that nothing was done to them before proceeding with the four hours' forced draft trials, which were successfully made with each boiler singly, 1,895 and 2,039 indicated horse-power being obtained with the starboard and port boilers respectively with 2.85 and 2.9 inches of air pressure, as against 2,250 indicated horse-power, the specified power per boiler. On examination, slight unimportant leakage of the tubes was observed, and the ferrules were practically free from oxide scale.

Subsequently, the eight-hours', natural-draft trial was repeated, 3,013 indicated horse-power being obtained with a mean air pressure of 1.05 inches. On examination, a few tubes were

found to be leaking slightly under the water pressure, but not sufficiently to necessitate further rolling. No difference was observed in the behavior of the iron and steel tubes.

Shortly after the above ferrule was suggested, Mr. Peck, of Messrs. Yarrow's firm, sent me a letter proposing a ferrule of the shape shown in Fig. 20. The points claimed for this, in his own words, were as follows:

"Something in this direction has, I am aware, been already proposed, but in this proposal you will see that the tube ferrule or protector does not touch the tube where it is fixed to the tube-plate, but is in contact with the tube only at a part where all its heat may be readily absorbed. The space between the protector and the tube is rather exaggerated in the sketch, but is only intended to be that due to expanding the ordinary straight tube."

It will be seen from this that Mr. Peck considered the space produced by the rolling of the tube in the tubeplate would be effective in producing the desired result, and it was not proposed to protect the tubeplate by any flanging of the ferrule.

The idea of a space formed between the ferrule and the tube at its junction with the firebox tubeplate was recognized as an important point, and a few hours subsequently Mr. Oram, Engineer Inspector, proposed to me the cap ferrule shown in Fig. 19, as one which would provide an effective air space between the ferrule and the tube at, and for a short distance beyond, its junction with the tubeplate, and by its cap also protect the greater portion of the firebox tubeplate from direct contact with the products of combustion, and proportionately reduce the formation of steam on the water side of the tubeplate. Experience has resulted in the shape of ferrule shown in Fig. 21, but it will be seen that the effective points remain the same.

In order to make a practical test, these cap ferrules were fitted in the *Barracouta's* boilers, which were of the double-ended, common-combustion-chamber type. The port boiler was fitted with wrought-steel ferrules, and the starboard boiler partly with the same and partly with malleable cast-iron ones.

On an eight hours' trial of the port boiler at natural draft, 978 indicated horse-power was developed with a mean air press-

ure of 1.07 in., as against 950 indicated horse-power specified, the trial being in all respects satisfactory. It was found that not a single tube leaked; some of the ferrules were loose and could be turned round by hand, but none of them could be swept out by a brush. The faces of the ferrules, and in a few cases the insides, showed signs of burning and scaling.

The loose ferrules were tightened by light rolling with expanders, and, on a four hours' trial of the same boiler at forced draft, 1,450 indicated horse-power was developed with a mean air pressure of 2.41 inches. Slight priming occurred, which prevented the development of the full power of 1,500 indicated horse-power. This trial was also satisfactory, examination showing that no tubes had leaked. The scaling of the ferrules seemed to be no worse, the first formation appearing in a measure to protect the ferrules from further burning. Next, an eight hours' natural draft trial was made of both boilers; 1,912 indicated horse-power was satisfactorily obtained with a mean air pressure of .69 inches, the specified power being 1,900 indicated horse-power. All the tubes in the port boiler were tight, but one tube in the starboard boiler showed slight indications of having leaked.

The scaling of the ferrules—more especially the wrought-steel ones in the port boilers—after this trial was observed to have increased; the first formations had in many cases curled up and broken off, leaving fresh surfaces exposed to the fire, and consequently new scalings had formed. A few ferrules were loose.

A forced draft trial was next made with the starboad boiler. Difficulties arose with the fans, and it was decided to stop the trial after three hours; 1,416 indicated horse-power was obtained with a mean air pressure of 2.3 inches. Not a single tube leaked.

Subsequently an eight hours' natural draft trial with both boilers was repeated, 1,932 indicated horse-power being developed with a mean air pressure of .77 inch. Examination of the boilers showed that not any tubes had leaked, a few ferrules were loose, and further breaking off of the scale and deeper burning of the ferrules had occurred. It was noticed that the scaling of the

malleable cast-iron ferrules appeared to be much less than that of the wrought steel.

At the conclusion of these trials the full water pressure test of 245 pounds was applied to both boilers, and not a single tube leaked.

After these satisfactory trials, the ferrules were fitted to the six furnace, double-ended, common-combustion-chamber boilers of H.M.S. *Thunderer*, which had given great trouble from leaky tubes at all powers down to less than one-third natural draft. On an eight hours' natural draft trial, after the cap ferrules were fitted, the indicated horse-power obtained was 5,900, with an average air pressure of 1.2 inches, the specified power being 5,500. Examination showed a few throat seams, rivets, and stay nuts to be leaking, but not one tube.

Next, a few hours' forced draft trial was carried out, the results being 7,066 indicated horse-power with an air pressure of 1.96 inches, that specified being 7,000. Examination showed only ten tubes leaking slightly out of about 3,000. With a view to testing the durability of the cap ferrules, the *Thunderer* was directed to proceed to Madeira and back at four-fifths of her specified natural draft power, *i. e.*, 4,400 indicated horse-power, and sailed for that purpose on July 16th at 6 A. M., arriving at Funchal at 1.30 P. M. on the 20th. Leaving again at 9.30 A. M. on the 23d, the power on entering the Bay was increased to 4,700 indicated horse power, and maintained till Portsmouth was reached at 3.45 P. M. on the 27th. The average speed out and home was 12.8 knots. Examination of the boilers showed one ferrule to be missing and its corresponding tube leaking; also one adjacent tube showed signs of having leaked slightly. Several ferrules had become loose, but could not be removed by hand. The ferrules were thickly furred in the manner frequently observed in torpedo-boat locomotive boilers, and about 30 out of 3,000 were worn out.

During the past week, the *Vulcan*, after having the vertical rows of tubes—those which had been removed to improve the circulation—replaced, and the tubes cap ferruled, has gone through a satisfactory four hours' forced draft trial, the indicated horse-

power being 12,032, as compared with 12,000 indicated horse-power specified, the mean air pressure being 1.8 in. These cap ferrules have been fitted to several other ships, having various types of boilers, with satisfactory results, and requests for them are being made by ships of the fleet with the view of protecting the tubeplates and ends from overheating produced by accumulation of grease or scale in the boilers, and they will be probably fitted to most vessels. In respect of protecting the tubeplates and ends from overheating, whether by want of circulation, excessive temperature in the combustion chamber, or from the presence of grease or solid matter, it is submitted these cap ferrules have fully answered their intended purpose.

In conclusion, I beg to take this opportunity of expressing the very deep obligation I feel under to my brother engineers of the Admiralty, Dockyard and Fleet, for their cordial co-operation and assistance in the endeavors that have been made to elucidate and overcome the diffiulties that have arisen in modern practice with marine boilers, and for the cool pluck and determination with which they have carried out their arduous duties on the steam trials in connection therewith. Last, but not least, a meed of praise is also, I consider, due to the artificers and stokers of the Fleet for their important share in this work.

Discussion.

(Reported in "Engineering.")

Mr. Yarrow, the first speaker, pointed out that the paper just read, as might have been expected from Mr. Durston, was one of the most interesting and instructive that had been laid before the Institution, and would form a most valuable addition to the Transactions. It would be admitted that when engineers found themselves faced by a difficulty which ordinary means do not enable them to overcome, it is by such a series of experiments as Mr. Durston had organized that light is thrown upon the subject. At the present time, therefore, his paper was of more than special value.

The experiments throughout proved conclusively the great difference in temperature which exists between the two sides of the

tube plate, and it was reasonable to suppose that the overheating of the side of the tube plate next to the fire, and the difference of temperature between the two sides of it, had been the chief cause of the trouble lately experienced with leaky tubes. The great extent of this difference of temperature is clearly shown, the speaker said, by the figures Mr. Durston had put before the meeting.

He understood that all the experiments tried had been made with steel or iron tube plates, and he would like to ask Mr. Durston if he had tried any similar experiments with tube plates of copper, a material which conducts heat six times as rapidly as steel. Consequently there would be a much less difference of temperature between the two sides of the plate, and the side next the fire would be maintained at a much lower temperature than would be the case if steel were the material selected.

Mr. Yarrow would also like to inquire whether any similar experiments had been made with tube plates less in thickness than described in the paper, as he thought that if steel tube plates of, say, $\frac{7}{16}$ inch in thickness, or even $\frac{3}{8}$ inch, were tested in a similar way to the manner adopted with the thicker plates, the difference of temperature between the two sides would be found to be very much reduced on account of the reduced distance the heat would have to travel.

He maintained, therefore, that the use of copper or thin steel tube plates might go far to diminish the difference of temperature between the two sides and to minimize the difficulties to be surmounted. If such experiments had not already been tried (and Mr. Durston, the speaker pointed out, said that the investigations are not yet complete, but would be carried further), he thought the Institution would be under a still further obligation to the author if such experiments with copper plates and thinner plates of steel were made, and the results laid before some future meeting.

The author had stated that copper or brass tubes do not stand so well as iron or steel tubes. This is what Mr. Yarrow would have expected, considering that the copper and brass tubes referred to were fitted into a steel tube plate; but it did not prove

that copper or brass tubes are inferior to steel when used in conjunction with copper tube plates, according to the practice of nearly all locomotive engineers in this country, and it was a combination Mr. Yarrow himself much preferred. If Mr. Durston had made any experiments indicating the behavior of copper or brass tubes in copper tube plates, the speaker thought it would no doubt be very interesting to the members of the Institution to know the results.

With regard to the Admiralty ferrules fitted to the *Barracouta* boilers, it appeared "some of them were loose, and could be turned round by hand. The faces of the ferrules, and in a few cases the insides, showed signs of burning and scaling." This was the result of an eight-hours' trial with only natural draft. Further on, the paper stated, with another "eight-hours' natural draft trial new scalings were formed and a few ferrules were loose." Again, it was said that another "eight-hours' natural draft trial was made and a few ferrules were loose, and further breaking off of the scale and deeper burning of the ferrules had occurred." Mr. Yarrow would like to ask Mr. Durston whether, with a continuous trial under forced draft, he did not think the ferrules would become still looser and show more signs of deterioration, and whether they would not be so far loosened as to be swept out by a brush, and ultimately become so deteriorated as to be readily completely burned away. His firm had found, under forced draft, a very large proportion of ferrules become so far loosened as to be easily swept out when cleaning the tubes, and, if the ferrules project much beyond the tube plate, their life is limited to a very few hours of continuous full-speed steaming. The importance of ferrules remaining in their place was clearly demonstrated, for the author had stated, in the case of the *Thunderer*, the result of "one ferrule being missing was to cause its corresponding tube to leak." Mr. Durston had also stated that "the ferrules were thickly furred in a manner frequently observed in torpedo-boat locomotive boilers, and about 30 out of 3,000 were worn out." Considering that this took place with a run when only four-fifths of the natural draft was adopted, it would point to a very serious furring up of the tubes and a large in-

crease in the number that would be worn out, if forced draft had been tried.

Continuing his observations, Mr. Yarrow said that the aim of the marine engineer in these days is to obtain the greatest power from the smallest weight of boiler, and the various experiments which Mr. Durston had described referred to a rate of combustion considerably below that which those who endeavored to obtain the lightest practicable boilers were aiming at. Mr. Durston had undoubtedly proved that the *Admiralty* ferrules are a great success, having enabled him to render available boilers which previously caused great trouble; but that did not prove that, with a further increase of the rate of combustion, with lighter and smaller boilers, the same success would be obtained. In fact, Mr. Yarrow said the results described pointed to exactly the opposite conclusion, because, with a greater rate of combustion and higher temperatures, such as is customary in torpedo-boat boilers, the experience of his firm with ferrules showed that they very rapidly deteriorate, that they constantly become loose, and many are pushed out by simply cleaning the tubes; and what is, perhaps, even worse, the small particles of coal adhere to that portion of the ferrule which is raised to a very high temperature, thus rapidly accumulating and closing up the orifice of the tube. This is a difficulty which all those who use a very high rate of combustion have found to be so far insurmountable, and the speaker could see no reason why the ferrules adopted by the *Admiralty* should be free from these difficulties. Owing to their large area of surface exposed to the heat without the means of conducting it rapidly away, he thought it reasonable to suppose these new ferrules would be placed in a more disadvantageous position than ordinary ferrules.

It was the speaker's opinion that we must look to other means of getting over the leaky tube difficulty in cases where the rate of combustion is greatly increased, thereby enabling boilers of smaller size and less weight to be available and to be capable of being forced to a much greater extent in cases of emergency. Mr. Yarrow suggested that, as the aim of the ferrule was to protect the tubeplate from becoming overheated, instead of

using ferrules having circular caps, leaving a portion of the tubeplate still exposed to heat, similar ferrules should be tried with a square face, so that the edges of the ferrules might come closer together and protect a still greater portion of the tubeplate.

As regards loosening of the ferrules, and the risk of their coming out, which the speaker felt sure would frequently arise with the Admiralty ferrules, he would suggest that they should be swelled out slightly at their extreme inner extremity by a drift or expander, so as to force themselves, as it were, slightly into the tubes. This would prevent the risk of their being pushed out by the brush when cleaning. For this purpose tubes reduced at the tubeplate might be used, or simply by reducing the internal diameter of the ferrule at its inner extremity and driving a drift to slightly open it out. The ferrules at this part might have some slits cut in them so as to be more easily expanded, and yet not offer any excessive resistance to being driven out when it was necessary to replace them.

In conclusion, Mr. Yarrow said that everyone must congratulate Mr. Durston and Mr. Oram on the measure of success which has been obtained by the Admiralty ferrule. It was a device which undoubtedly so far enabled boilers to be available which previously could not be worked; but he thought marine engineers should endeavor in the case of new boilers to design them so that the difficulties experienced were got over by means not open to the objections he had indicated, and until this was done, he felt sure Mr. Durston would agree that the problem could not be looked upon as solved.

Mr. Stromeyer remarked that experiments similar to those mentioned in the early part of the paper had been carried out by Mr. Hirst, and in one sense they were even more valuable than those mentioned to-day, because he measured not only the temperature of the plate, but also the rate of evaporation, and found that this had a great influence. Unfortunately, he only experimented on iron plates, whereas, now, steel was universally adopted for boilers.

Mr. Stromeyer then asked for further information on various points, and pointed out how necessary it was to know exactly how much air was passing through the tubes of a boiler the temperatures of which were being measured.

In discussing the tubeplate troubles, Mr. Stromeyer mentioned some experiments which he had recently witnessed, and which seemed to point very clearly to the real cause of leaky tubes, and possibly to a remedy for this state of things. It did not seem to be generally known, though it was almost self evident, that when annealing iron bars or plates, which, in the process of manufacture, must have been stretched, they tend to revert to their original condition and grow permanently shorter. Similarly, if steel bars or plates are first shortened while cold, they will, on annealing, increase in length.

These facts were very forcibly demonstrated in the experiments which the speaker had witnessed. Two holes were bored into a dished plate; an ordinary tube was expanded into the one hole, and another, whose end had been first compressed, was inserted into the other. After a very severe trial over a smith's fire, extending over several hours, and embracing such treatments as evaporating 100 lbs. of water per square foot per hour, raising the plate to a blue heat and quenching in water, and then heating it to such a temperature that water which was repeatedly dashed on it remained in a spherodal condition, the tubes were suddenly found to be slack, the common tube being quite loose. On carefully remeasuring its internal diameter, it was found to have contracted $\frac{1}{40}$ in., which was quite sufficient for the most serious leakage. The compressed tube, on the other hand, had expanded where possible, viz., the projecting end of the tube had grown larger than the hole through which it had passed, but it only touched the fire side edge of the tube hole, for on the water side its diameter had increased more than the diameter of the tube; the hole had, in fact, grown more taper than it was before.

Mr. Langton, of the firm of Humphrys, Tennant and Co., said that Mr. Durston had referred to the screwed ferrule intro-

duced by his firm, and had also made reference to the trouble that had been experienced in the *Medea*. In that ship, it should be explained, the tubes had been rolled and re-rolled, so that the success of the ferrule which, however, should be described as a cap rather than a ferrule, was more creditable. There was a falling in the indicated horse-power, and that was because of the diminution of the heating surface on account of the abstraction of the tube. Reference had been made to the difficulty of withdrawing the screwed cap or ferrule, but that was not so great as was supposed; and in the case of the *Medea* it had been done very easily by means of a key constructed for the purpose. The essential feature about this ferrule was the metallic contact that was maintained, which prevented burning away.

Mr. McFarlane Gray said that the ferrules were likely to get knocked out. He would suggest a cheap and effective means of keeping them in their place. This he illustrated by a sketch on the blackboard, and which turned out to be a false tube plate placed in front of the ordinary tube plate, the whole corresponding with the tubes, but somewhat smaller.

Mr. John Scott, of Greenock, said there were several lessons, apart from the protection of tube ends, to be learned from this very valuable paper. Firstly, they were the first quantitative experiments yet made as to the effect of mineral oil in a boiler. They all knew that there was an effect, but they were not aware before how sensible it was in regard to mineral oil in solution, and what he described as a thick or "claggy" state. He would confirm what Professor Lewes had said two or three years ago as to the efficiency of tube surface. The diagram Mr. Durston had put on the wall, showing the varying temperatures of different lengths of the tube, reminded him of the experiments of Mr. Wye Williams, who had divided his boiler into different lengths and found the evaporation which took part in the respective divisions, this course only being open to him, as he had not the Le Chatelier pyrometer. The results then obtained corresponded with those shown by Mr. Durston. Mr. Durston's experiments also showed the greater value of furnace surface compared to

tube surface. In conclusion, Mr. Scott would congratulate the author and his colleagues on solving a difficult problem which had given them so much trouble during a long time past. He was sure the public would be glad to learn that the difficulty was at an end.

Admiral Long, as a naval officer of the executive branch, could not discuss engineering questions, but he felt that he must congratulate Mr. Durston and his staff on solving a difficult problem. To him it seemed that the ideal warship should be able to catch its enemies, and beat them when caught. To do this, the most important link was in the engine-room.

Mr. Seaton, of Hull, pointed out that the question of stoking athwartships had been raised. For his own part he did not say how much there might be in this point, but it was worth calling attention to; for, as shipowners said, their troubles began with boilers when they were placed fore-and-aft. So firmly was this idea fixed in the minds of some, that they refused to have their boilers fixed in that position. That was with regard to priming chiefly, and the speaker had often thought that the absence of priming in the locomotive boiler was due to the shaking up it got on the journey. If the boilers were placed athwartships, the tubeplates were washed, if there were any motion of the sea at all.

Mr. Seaton wished to add his testimony to that which had already been borne as to the credit due to Mr. Durston and Mr. Oram for the able manner in which they had got through their difficulties.

It was no great mystery why tubeplates got hot. Evaporation depended on the difference in temperature on the different sides of the heating surface. Mr. Seaton was quite at one with Mr. Yarrow on the efficiency and desirability of using thin tubeplates. Thick tubeplates were first forced on the marine engineer by the Board of Trade, and he thought they were at the bottom of all our troubles. If we could have thin tubeplates, the trouble would vanish. He thought there was no doubt that the day was fast coming when shell boilers would be a thing of the past. The tube boiler was in the ascendant, and would soon be with us,

especially for the high pressures demanded by quadruple-expansion engines. The French were ahead of us in this matter, having applied tubular boilers on a large scale, and some form of water-tube boiler the speaker felt convinced would have to be adopted.

Mr. Macfarlane Gray suggested that Mr. Wye Williams' diagram as to the evaporative efficiency of the different lengths of tubes should be added to that of Mr. Durston, now shown on the wall, when the paper appeared in the transactions.

Mr. Joy desired to supplement Mr. Seaton's remarks by the statement that in the United States also they were ahead of us in regard to pipe boilers. In the past they had had trouble in this country with the pipe boiler, but he thought that with increased knowledge, improved appliances, and the better engineering practice of the present day, the old difficulties would be overcome. They arose from want of circulation. That was a thing in which great advances had been made recently. Mr. Joy next referred to certain steamers which have lately been built in the United States, and in which he was interested, and of which he gave a few particulars. Mr. Joy evidently referred to some vessels now building for lake service in America. These vessels are of remarkable interest. Two are being constructed now, and four others are, we believe, to follow. They are to run from Buffalo to Duluth, a distance of 1,110 miles, the passage occupying about 60 hours, including stoppages. The length of these ships is 360 feet between perpendiculars, 44 feet beam, and 34 feet deep. They are to have quadruple-expansion engines, driving twin screws. The indicated horse-power will be 7,000. The cylinders are to be 25 inches, 36 inches, 51½ inches and 74 inches, the stroke being 34 inches. They are to have Joy valve gear; the steam is to be generated by 28 Belleville boilers, the pressure being 195 pounds. The speed is to be 20 knots.

Mr. Howden, of Glasgow, was the next speaker. He referred to the ferrules which had been adopted by the Admiralty. He looked on them only as a temporary remedy for a radical evil. He did not see why they should be wanted at all, if

boilers were properly designed and worked. He had had much experience in forced draft, and had never had a drop of water out of the tube ends. With regard to leakage being due solely to temperature on the tubeplates, he held that it was the change of temperature that caused the evil. The opening of the furthest door, when working under forced draft with the closed stokehold, caused a rush of cold air to be admitted to the furnace, and this striking against the tubeplate and tube ends caused the difficulty. There were boilers working with forced draft on his system on the Atlantic with a ratio of 22 to 23 horse-power per square foot of grate surface, and there was never trouble from leakage at the tube ends. But in that case cold air did not rush in and strike against the tubeplate. The speaker held that it did not matter what the temperature might be against the tubeplate so long as the latter was not suddenly cooled. He had been advancing these opinions for several years, and, in spite of that, they still saw these elaborate appliances adopted by the admiralty in order to overcome an evil which should not exist.

Mr. Durston, in replying to the discussion, said he was obliged to Mr. Yarrow and the other speakers for their criticisms on his paper and for the suggestions they had made. His object in writing a paper was to get from brother engineers expressions of opinion on this subject. He hoped to try copper tubeplates and also thinner tubeplates; in fact, he was working in that direction. With regard to the wear of the ferrules, in the early days they wanted to try them as soon as possible, and, therefore, they took the material ready to hand. That was steel tubes which were in stock in the dockyard and which were worked up into ferrules. In doing this they were thinned at the ends, which was not desirable. Afterwards they had made a number of malleable cast-iron ferrules, and that was the material they now used. [We believe that it is the custom to supply vessels with one set of spare ferrules for a commission—that is to say, one set fixed and one set spare for the period of three years.]

With regard to the ferrules getting loose and being knocked out, they did not find difficulty in that direction. The ordinary

ferrule was more likely to become loose than these. He had great pleasure in testifying to the value of Mr. Peck's suggestion, and which had had such successful results, and he considered that the *Admiralty* officials—that is, Mr. Oram—had extended the suggestion. With regard to swelling out the ferrule, as Mr. Yarrow had suggested, in order to keep it in the tube, he thought that it would make an undue resistance. They had just concluded the trials of the *Vulcan*, where they had got 22 horse-power per square foot of grate on the forced-draft trial, and the run had proved quite satisfactory. He had no record of the velocity of the air.

As to the cost, the ferrules were put down at sixpence each. He had not tried a false tubeplate, as suggested by Mr. Macfarlane Gray. Mr. Scott had made reference to the work done by the staff of contractors. Mr. Durston fully recognized the valuable aid that was rendered by the contractors and those they employed, as did all Admiralty officials.

Speaking of the position of boilers, Mr. Durston pointed out that the boilers of the *Thunderer* were athwartship, while those of the *Vulcan* were placed longitudinally. He agreed that the course of invention in boilers was tending towards the tubulous boiler. The Admiralty were now trying experiments both with foreign and British designs, and he thought in this respect that the British design would again come to the front.

He did not agree with what Mr. Howden had said in regard to the rush of cold air on the tubeplate being solely responsible for the bad effect produced. That was not borne out by the experiments which he had quoted in his paper, in which the fan was kept going after the fire had been withdrawn. In that case cold air was brought against the tubeplate, and yet there was no leaking. He was of opinion that the rush of cold air would not cause damage if the tubes and tubeplates were not first overheated.

DUTY TRIAL OF A BLAKE HIGH DUTY PUMPING ENGINE, AT THE CITY WATER WORKS, NEWTON, MASS.*

By F. W. Dean, M. E.

The contract required an engine capable of pumping 5,000,000 gallons in twenty-four hours, with a guaranteed duty of 115,000,000 foot pounds of work by the consumption of a quantity of water evaporated into dry steam, the equivalent of which is 1,100 pounds of water evaporated from and at 212° Fahrenheit. The method of computing this duty is as follows: "The area of the plunger (*s*) in square feet will be multiplied by the pressure in pounds due to the total lift in feet, which will be obtained from the average reading of a correct pressure gauge attached to the force main near its connection to the pump, to which will be added the pressure due to the difference of level between the zero of the gauge and the water in the pump well, the product thus obtained will be multiplied by the total number of feet traveled by the plungers on their discharging strokes. This product equals the total work in foot-pounds. The amount of water evaporated by the boilers during the trial will be reduced to an equivalent number of pounds of water evaporated from and at 212° Fahrenheit. The total work in foot-pounds divided by this equivalent evaporation and the quotient multiplied by 1,100 will be the duty of the engine in foot-pounds per specified unit of water evaporated into steam."

Soon after the contract was made, the American Society of Mechanical Engineers advocated, through a committee instructed to devise a standard method of conducting duty trials

* Extract from Official Report to the Water Board of the city of Newton, Mass., December 27, 1892.

of pumping engines, the computation of duty upon the consumption by the engine of 1,000,000 unites of heat. In consequence of this, the duty has been computed upon this basis as well as upon the other.

The contract also embraces the installation of two boilers of the locomotive type having the Belpaire system of staying the crown sheets. Each boiler was to be sufficiently large to easily furnish steam for the engine at full power. Through the liberalty of the contractors, these boilers were designed to safely carry 150 pounds by gauge to the square inch, although the usual working pressure was to be less than 130 pounds. These boilers were guaranteed to evaporate no less than eleven pounds of water per pound of combustible reduced to the equivalent of feed water at 212° and evaporation at that temperature.

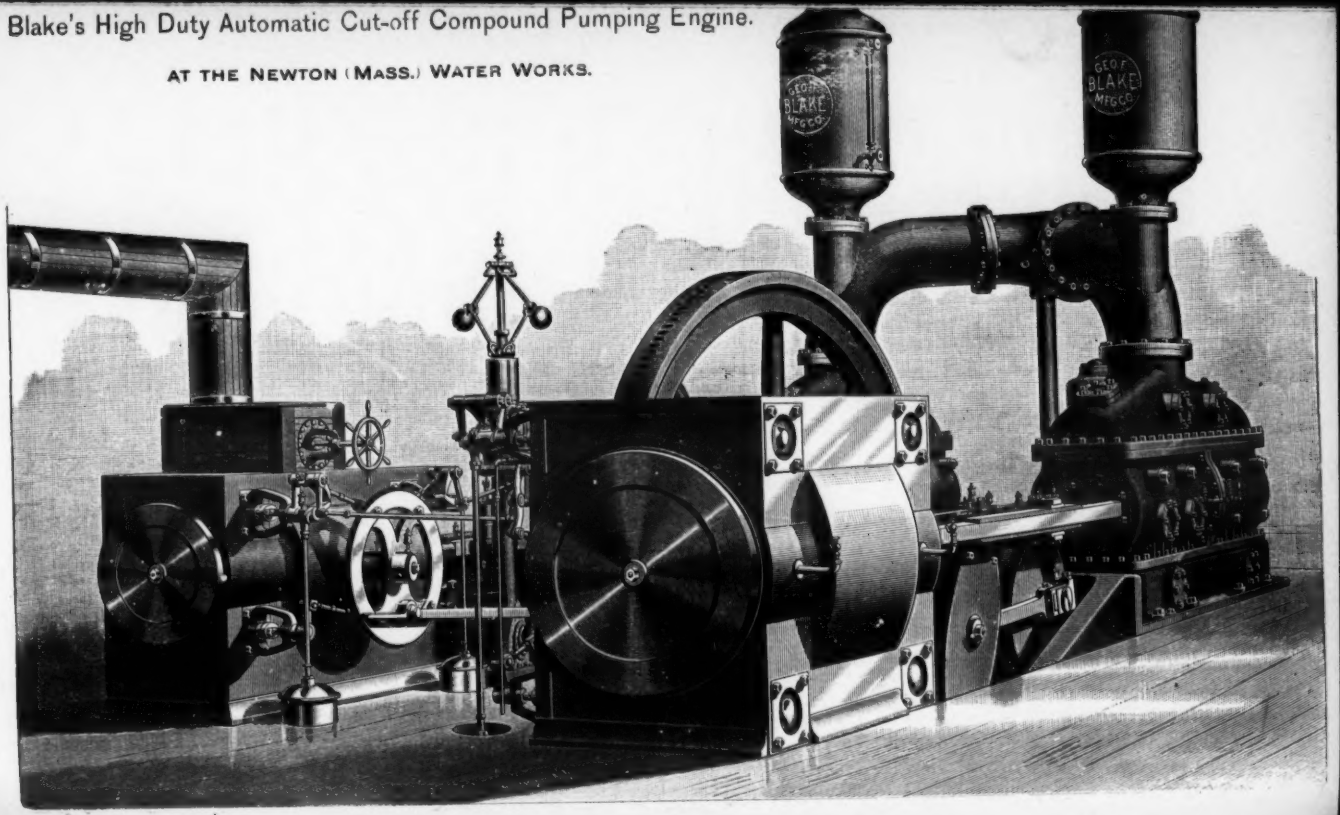
The engine is a new type of cross-compound fly-wheel horizontal-pumping engine, with the Corliss type of valve gear, designed and built by the contractors, the Geo. F. Blake Mfg. Co. It consists of one high-pressure cylinder and one low-pressure cylinder, side by side, and two double-acting horizontal pumps having large valve area. Each pump has its axis coincident with that of one steam cylinder, and the companion steam piston and pump rods are both keyed to a crosshead about midway between the steam and pump ends of the engine, there being therefore two crossheads. These crossheads slide upon guides which form tie-rods between the steam cylinders and pumps.

Near each steam cylinder there is a vertical beam working on horizontal bearings formed in a lower bed plate. These beams are operated by links passing from journals on the crossheads to pins in the upper ends of the beams, and motion is transmitted to the fly-wheel shaft by means of connecting rods passing from pins near the middle of the beams to the pins of cranks on the ends of the shaft. The shaft rotates in boxes formed in the bed plate before mentioned. The cranks are at right angles to each other, the low-pressure crank leading.

The valve gear possesses novel features. Among them may be mentioned the absence of springs from the latching mechanism, and the operation of this mechanism by a special eccentric

Blake's High Duty Automatic Cut-off Compound Pumping Engine.

AT THE NEWTON (MASS.) WATER WORKS.



for each cylinder. There is a high-speed centrifugal governor which varies the cut-off through a wide range. The inlet and exhaust valves of the low-pressure cylinder are driven by different eccentrics.

The cylinders are jacketed by steam on its way to the steam chests, and between the cylinders there is a tubular reheater, the reheating being done by live steam taken from the main steam pipe near the engine.

There is a coil feed-water heater in the exhaust pipe of the low-pressure cylinder, and the main feed pump forces the water through this coil to the boiler. Another pump forces the water formed by the condensation of the live steam in the reheater direct to the boiler. The condensation in the main steam pipe, that in each jacket and that in the exhaust steam side of the reheater, are trapped and wasted.

The air pump is vertical and single acting. This pump and the feed pumps are operated from arms on an extension of the low-pressure beam axis pin.

THE OFFICIAL TRIAL.

The fires in No. 2 boiler, which had been thoroughly cleaned, were lighted some four hours before the trial was to begin. The engine was run also for about an hour before the trial, thus heating both boiler and engine to their normal working temperatures. The engine was stopped and the fires were drawn. New fires were immediately started, and the pressure of steam and height of the water accurately taken. As soon as practicable the engine was started, and the boiler test was considered as begun, the time being 9.42 A. M., September 30, 1892.

The feed water weighing also began, the reheater drain was running to waste, and each man was at his post. At precisely 10 A. M. the feed water level in the lower tank and the water level in the boiler were accurately noted and the reheater drain turning into a hogshead of cold water placed upon scales. Previous to beginning the boiler test, all pipes in both boiler and engine rooms which could convey either water or steam to or from any point so as to vitiate the accuracy of the results were

disconnected, and it was ascertained that no valves leaked. In connection with this, it may be stated that all pressures were taken with nicely-tested gauges which were known to be correct, all temperatures with high-grade tested thermometers, and all indicators were provided with tested springs. All scales used for weighing coal and water were made correct with standard weights.

The pressure of the water in the force main was taken every ten minutes and the distance below the point at which this pressure existed was measured by an adjusted scale resting upon a float upon the water in the pump well. It was read simultaneously with the pressure gauge. The scale also showed the total suction lift of the pump, and this was read every ten minutes. The total pressure against which the pump acted was computed from the gauge and float readings and the weight of water corrected for its temperature.

Indicator diagrams were taken from the steam cylinders every twenty minutes, two Crosby indicators being used on the high pressure cylinder and two Thompson indicators on the low pressure cylinder. Indicator diagrams were occasionally taken from the pumps with Thompson indicators.

Pressures were taken every twenty minutes at the boiler, at the throttle, and on the steam supply end of the reheater. The counter was read every twenty minutes. Temperatures were taken every twenty minutes of the feed water before passing the heater and immediately after passing this heater; of the reheater drain immediately after leaving the reheater and about four feet beyond; of the feed water where it entered the boiler, and of the flue temperatures during the first nine hours and the last two hours of the test. Only one reading was taken at the hot-well.

No calorimetric observations of the quality of the steam were taken, as this was done by Professor Peabody a few months before, his results being accepted as applicable to this test. The calorimeter was applied near the throttle, and the moisture was found to be $\frac{7}{10}$ of one per cent. This was held to be due to condensation in the steam drum and steam pipes, which were rather long. The boiler was supposed to furnish dry steam.

As the time approached for ending the test, the fire was burned down as low as practicable while keeping the pressure nearly constant and the water in the boiler at its normal height. At the end of twenty-four hours, the height of the feed water in the lower barrel was marked, the water in the boiler being at the original height, and the jacket drain turned to waste. The engine test was then completed. The engine was immediately stopped, the water brought to the original boiler test level, with the original pressure and the boiler test ended. The fire was then drawn and the unconsumed coke picked out and deducted from the coal fed into the fire boxes. The level of the feed water in the lower tank was noted at the end of the boiler test.

After the test was over, the quantities of water necessary to bring the water in the lower feed tank to the original engine test and boiler test levels were weighed.

It is obvious that the engine test was shorter than that of the boiler, and the coal quantity to be credited to the engine was ascertained by making it proportional to the number of heat units which passed to the engine from the boiler during the engine test.

In the table which follows, it will be seen that the amount of steam used per hour per indicated horse-power is given with the moisture deducted in order to compare this result with similar results from other engines. While it is not strictly proper to make this deduction, the result is probably more nearly correct than if the deduction were not made. It is true that the heat in the moisture passes through the engine and may do good somewhere, but, considering the fact that the moisture is known to do harm, and that the engine is not responsible for it, the benefit from the heat is doubtful. Little or none of it can flash into steam during its course because heat of vaporization is so much greater than heat of liquid. This discussion seems necessary, in view of the fact that the steam consumptions given place this pumping engine among the most economical in the world.

The following tables give the principal dimensions of the engine and boilers, with the results of the trial in detail:

DIMENSIONS OF THE PUMPING ENGINE.

Type: Cross-Compound, Fly-wheel, Automatic Valve Gear.

Diameter of high pressure cylinder	21 inches.
Diameter of low pressure cylinder	42 inches.
Diameter of each pump plunger	13½ inches.
Diameter of piston and plunger rods	4 inches.
Stroke of all pistons and plungers	40 inches.
Diameter of one single-acting air-pump	26 inches.
Stroke of same	12 inches.
Diameter of single-acting plunger feed pump	4½ inches.
Stroke of same	6 inches.
Diameter of pump plunger for returning reheater drain to boiler	2 inches.
Stroke of same	6 inches.

DIMENSIONS OF THE BOILER.

Smallest inside diameter of shell	84 inches.
Inside length of fire-box	8 feet.
Inside width of fire box (twice 3 feet 5½ inches)	6 feet 10½ inches.
Length of combustion chamber	4 feet.
Length of tubes	16 feet.
Diameter of tubes, outside	3 inches.
Number of tubes	170.
Length of grate	7 feet.
Width of grate (twice 3 feet 5½ inches)	6 feet 10½ inches.
Heating surface	2,310 square feet.
Grate surface	48.125 feet.
Ratio of grate to heating surface	48.00.

ENGINE TEST.

Duration of trial	24 hours.

AVERAGE PRESSURES, ETC.

Steam pressure at the boiler, by gauge	125.60 pounds.
Steam pressure at the throttle valve, by gauge	122 31 pounds.
Pressure of atmosphere, by barometer	14.72 pounds.
Absolute steam pressure at boiler	140 32 pounds.
Absolute steam pressure at throttle valve	137.03 pounds.
Vacuum, by gauge	27 inches.
Weight of one cubic foot of water at 59.5 degrees F.	62.37 pounds.
Water pressure in gauge chamber	96.83 pounds.
Difference in level of gauge and water in pump well	21.14 feet.
Corresponding pressure	9 156 pounds.
Total water pressure (head)	105.986 pounds.
Minimum suction lift	21.84 feet.
Maximum suction lift	23 23 feet.

AVERAGE TEMPERATURES.

Of engine room ..... 75.96 degrees F.
Of external air ..... 60.50 degrees F.
Of feed water at boiler, as deduced from preliminary test ..... 129.90 degrees F.
Of reheater drain at reheater ..... 341.40 degrees F.
Of reheater drain at boiler, as deduced from preliminary test ..... 280.30 degrees F.
Of water in pump well ..... 59.50 degrees F.

FEED WATER.

Total water pumped into boiler during engine test ..... 84,491 pounds.
Moisture in steam at throttle ..... 0.7 per cent.
Steam used in cylinders of engine ..... 79,437.12 pounds.
Steam used in reheater ..... 5,053.88 pounds.
Steam used per hour per indicated horse-power, excluding moisture ... 13.97 pounds.
Steam used per hour per indicated horse-power, including moisture .... 14.07 pounds.

BRITISH THERMAL UNITS.

Per pound of steam used in the cylinders 1,093.05
Per pound of steam used in the reheaters 870.55
Total used in cylinders and reheaters ..... 91,228,399.
Total used in cylinders and reheaters in 24 hours ..... 91,228,399.
Supplied by boiler per pound of steam from 129.9 degrees F. to 140.32 pounds absolute ..... 1,091.56.
Total supplied by boiler in 24 hours 92,215,632
Total supplied per pound of coal 10,816.8
Total supplied per minute per I.H.P 253.14
Efficiency of the engine ..... 16.9 per cent.

AVERAGE POWERS DEVELOPED.

Total number of revolutions in 24 hours ..... 56,326.
Average number of revolutions per minute 39.115
Average mean effective pressure in H.P. cyl ..... 42.44 pounds.
Average mean effective pressure in L.P. cyl ..... 12.50 pounds.
Horse-power developed by H.P. cyl 114.03
Horse power developed by L.P. cyl 136.24
Horse power developed by both cyls. 250.27
Percentage of total power developed in H.P. cyl ..... 45 6 per cent.
Percentage of total power developed in L.P. cyl ..... 54.4 per cent.
Horse power of water plungers 229.23
Efficiency of mechanism ..... 91.6 per cent.
Steam accounted for by indicator at cut off H.P. cyl ..... 8.78 pounds.
Steam accounted for by indicator at release H.P. cyl ..... 10.35 pounds.
Steam accounted for by indicator at cut off L.P. cyl ..... 9.20 pounds.
Steam accounted for by indicator at release L.P. cyl ..... 11.41 pounds.
Proportion of above to feed water consumption at cut off H.P. cyl ..... 63 per cent.
Proportion of above to feed water consumption at release H.P. cyl ..... 74 per cent.
Proportion of above to feed water consumption at cut off L.P. cyl ..... 66 per cent.

AVERAGE POWERS DEVELOPED.

Proportion of above to feed water consumption at release L.P. cyl..... 82 per cent.
Percentage of total steam used by cyls ..... 94.02 per cent.
Percentage of total steam used by reheater..... 5.98 per cent.

BOILER TEST.

Duration..... 24.38 hours.

AVERAGE PRESSURES.

Steam pressure at boiler, by gauge..... 125.60 pounds.
Atmospheric pressure, by barometer..... 14.72 pounds.
Absolute steam pressure ..... 140.32 pounds.
Force of draft, inches of water..... .35 inches.

AVERAGE TEMPERATURES.

Of external air..... 60.5 degrees F.
Of fire room..... 81.5 degrees F
Of escaping gases during the first nine hours..... 391 degrees F.
Of escaping gases during the last two hours..... 494 degrees F.
Of feed water before entering heater..... 59.5 degrees F.
Of feed water before entering boiler..... 129.9 degrees F.

FUEL.

Total weight of dry coal consumed, including equivalent of wood... 8,653.6 pounds.
Total dry refuse, weight..... 632.6 pounds.
Total dry refuse, percentage..... 7.3 per cent.
Total combustible weight ..... 8,022 pounds.
Total combustible percentage..... 92.7 per cent.
Total calorific value of one pound of coal by experiment..... 13,452 B.T.U.
Total calorific value of one pound of coal by analysis ..... 14,373 B.T.U.

CALORIMETRIC TEST OF STEAM.

Percentage of moisture in steam ..... $\frac{7}{10}$ of 1 per cent.

WATER.

Total weight of water pumped into boiler and apparently evaporated, 85,011 pounds.
Equivalent water evaporated from and at 212 degrees F..... 96,912 pounds.
Equivalent total heat derived from fuel in British thermal units..... 93,602,212.
Equivalent water evaporated from and at 212 degrees per hour..... 3,975 pounds.

ECONOMIC PERFORMANCE.

Water evaporated per pound of dry fuel from average temperature of feed (129.9 degrees) at average steam pressure (125.6 pounds)..... 9.91 pounds.
Equivalent water evaporated from and at 212 degrees..... 11.19 pounds.
Equivalent water evaporated from and at 212 degrees per pound of combustible..... 12.08 pounds.
Equivalent total heat derived from a pound of dry coal..... 10,816.8 B.T.U.
Dry coal burned per hour per square foot of grate surface..... 7.38 pounds.
Feed water evaporated per hour per square foot of heating surface from and at 212 degrees F..... 1.51 pounds.

DUTIES.

Based upon the heat given to each pound of water fed to the boiler.

Per Newton basis	125,497,323 ft. lbs.
Per 100 pounds of coal burned	127,779,469 ft. lbs.

Based upon the heat received and rejected by the engine.

Per 1,000,000 British thermal units	119,406,777 ft. lbs.

ANALYSIS OF COAL USED.

Kind of coal: George's Creek, Cumberland.

	Per cent.
Moisture	.58
Carbon	79 40
Hydrogen	5.11
Oxygen	3.80
Nitrogen	1.25
Ash	9.24
Volatile sulphur	.62

The duty on the Newton Basis is 125,497,323 foot pounds, or 9.12 per cent. above the guaranteed duty of 115,000,000 foot pounds.

The boiler evaporated 12.08 pounds of water per pound of combustible from and at 212 degrees F., and therefore exceeded the guarantee of 11 pounds by 10.98 per cent.

The engine runs smoothly at all times, and will always be very economical. Whether looked upon from the standpoint of excellence of design, workmanship, durability or stability, the engine is second to no horizontal pumping engine in the country.

The results of the boiler trial show the boiler to be among the most economical on record, and this high degree of economy should be maintained indefinitely. * * *

EFFECT OF AUXILIARIES ON ECONOMICAL SPEED.

ADDITIONAL DISCUSSION BY FREDERICK MERIAM WHEELER, ESQ., ASSOCIATE.

There is no doubt that much steam can be saved for the use of the main engines by reducing to a minimum the number of auxiliaries aboard ship, and improving their economy by adopting the highest economical types. For instance, one set of steam cylinders could be saved by combining the air and circulating pumps, and using compound steam cylinders for operating same. With this pump the exhaust steam can be carried to one of the receivers of the main engine, or to the condenser, as desired. In the former way it would be operating the pump practically on the triple-compound system. I have found by experience that, by the simple compounding of a steam pump, the consumption of steam can be reduced to considerably less than one-half. The same remark can also be applied to the main feed pumps, or any other pumps that are in continual service and of any considerable size.

Chief Engineer Isherwood, in his digest of the test of the engine of the Hoboken ferry-boat *Bergen* (published some years ago in the JOURNAL of the A. S. N. E.), showed that about twenty per cent. of all the steam generated was used by the auxiliaries of the boat, although the air-pump was connected to the main engine. The circulating pump, as also the fire and bilge pumps, were of the duplex system, and exceedingly extravagant in the use of steam. In other words, the pumps were indicating entirely too much power for the actual amount of work done, and too much steam per I.H.P. The auxiliaries of the new ferry boats of this line, the *Bremen* and *Hamburg* (according to Prof. Denton's report) only consumed ten per cent. of the steam generated

by the boilers. These boats have combined air and circulating pumps (Blake system), with compound steam cylinders, and are doing their work with less than 1.5 per cent. of the I.H.P. of the main engines, and with a consumption of steam of only 40 pounds (weight) per I.H.P. per hour.

In feeding boilers of modern engines, I would like to emphasize the fact that it costs more to feed boilers of triple-expansion engines than it does the boilers of compound engines; the work due to the increase of pressure in the case of the former is greater than the saving of steam of the former over the latter type of engine.* The feed pumps should therefore be arranged with compound steam cylinders.

By the way, Mr. Hollis, on the third page of his paper, where he enumerates the number of the main engine auxiliaries on board the armored cruiser *New York*, in mentioning the Blake vertical air pumps for the main condensers, speaks of them as being of the "duplex" type. This is a slight error. They are of the "twin cylinder," and not of the duplex type. I mention this because the latter type is proverbially more extravagant in the use of steam than the regular direct-acting system of steam pumps. As an illustration, where Mr. Isherwood mentions the duplex circulating pumps on the ferry boat *Bergen*, he quotes that it takes over 120 pounds (weight) of steam per I.H.P. per hour, while Chief Engineer George W. Baird refers to the single direct-acting circulating pump on the *Albatross* as not taking over 80 pounds (weight) steam per I.H.P. per hour. The latter gentleman also refers to the combined air and circulating pump on the *Dolphin*, where a horizontal double-acting circulating pump and two vertical single-acting air pumps are driven by a single direct-acting steam cylinder. Now, this combination pump (Blake system) on test did not take over 60 pounds (weight) of steam per I.H.P. per hour. Mr. Baird also shows that on the

* On a basis of 19 pounds weight of steam per I.H.P. per hour for compound engines, and 16¼ pounds for triple expansion, it will require for a given I.H.P. about 15 per cent. less power (theoretically) to feed the boilers of the former than of the latter, it being understood that 120 pounds of steam is carried on the boilers for the compound and 160 pounds on the boilers for the triple-expansion engines.

trial trips of the *Dolphin* (not including the first) the power of this pump averaged about 1.8 per cent. of the I.H.P. of the main engine. This amount can and has been reduced to less than 1.5 per cent. by later and more improved designs of this system of pumps.

It might be of interest to mention here that, from recently-conducted tests, it was shown that it requires very little more steam to work the air and circulating pumps, when operated together by one direct-acting steam cylinder, than it does to simply work an independent air pump alone, the difference being less than 15 per cent.; hence, my recommending the combination of the air and circulating pumps, where circumstances will permit of its use. The explanation of this is easily recognized, as the load of an air pump is not reached until the piston or buckets approach the end of the stroke. At this point we have the maximum momentum of the water and working parts of the circulating pump, so that the load of the air pump comes in very nicely at the very point of the stroke where we have the greatest power.

NOTES.

THE FLOATING DERRICK AT CRAMPS'.

The great floating derrick attached to the shipyard of the William Cramp & Sons' Ship and Engine Building Company is now in complete order and is running very successfully. Not long ago it lifted a 70-ton boiler off the dock, towed itself a distance of 150 feet, and lowered it into the cruiser *New York* in less than twenty-five minutes.

Only two men were required to operate the machine, and the ease with which the 70-ton bundle was lifted from one place to another suggested play rather than mechanical engineering of high grade.

The hull, which is, practically speaking, a strongly-built tank, is rectangular in shape and measures 73 feet long by 69 feet wide, with a depth of 13 feet 4 inches, and is divided into twenty distinct water-tight compartments by bulkheads, three of which run in the direction of the width and four intersecting same at right angles, running in the direction of the length. The superstructure, or tower, is mounted upon this hull a little forward of the intersection of the axis of its breadth and length, but in the medial line of the width—or, more correctly speaking, its long axis—for the purpose of giving the boom the greatest available overhang or reach.

The series of five compartments constituting the rear end of the pontoon can be independently filled with water and thereby act as counterballast when extra heavy lifting operations become imperative. These water-ballast tanks are filled and emptied by a high-pressure pump, having an 8-inch suction and 7-inch discharge. This pump is fitted up with all the essentials for extinguishing fires, and will prove a very effective auxiliary to the fire department of the yard. Each of the twenty compartments which constitute the hull is accessible.

The tower is built of twelve columns, made of two 12-inch channels placed back to back, resting on and bolted to the base piece, which is 40 feet in diameter and cast in twelve segments. At the top, the columns of the tower are bolted to the crown casting, through which the mast or king-post, 40 inches in diameter, passes to the step casting, 11 feet in diameter and 15 feet below the crown casting. The mast and its load are carried by 42 four-inch hardened steel balls, true to the thousandth part of an inch, which bear on interchangeable rings held by the step casting.

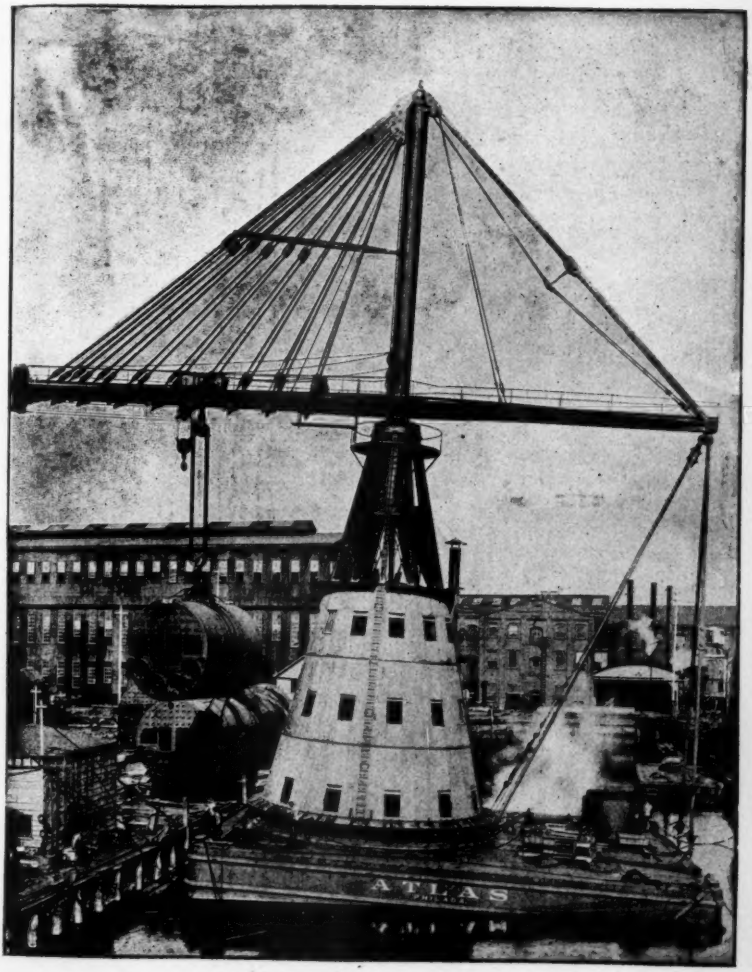
The step casting is tied to the crown or cap casting by twelve 1½-inch bolts, and to the columns by wrought iron brackets, thus making the pull direct on the columns of the tower. The tower is further strengthened by three hoops of 9 x 1 inch iron, bolted to the columns, and also by a series of rods and ties at all points of strain.

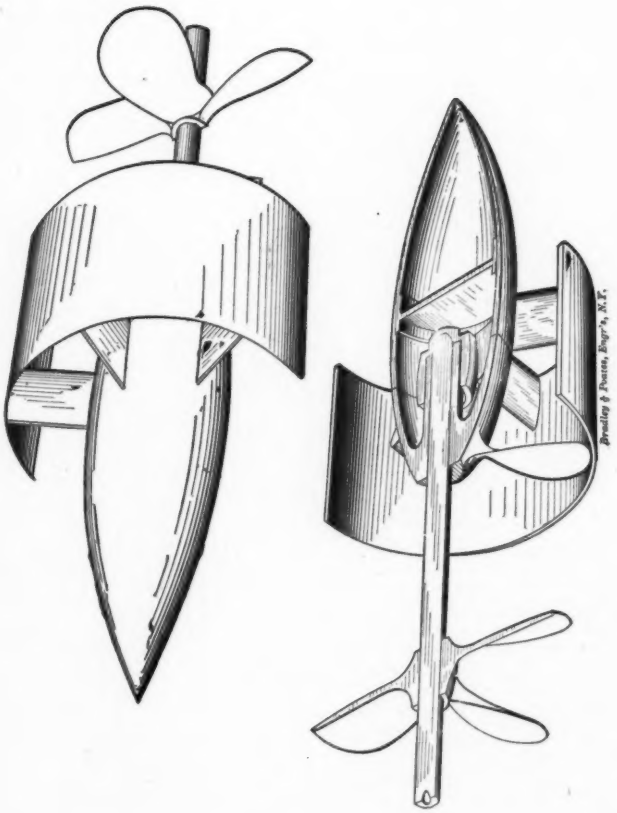
The front and back booms, which operate on the principle of yard-arms, are just above the crown casting and connected to it. The beam is 65 feet above deck and will have a calculated clear lift of 40 feet, the difference being taken up with the gear and attachments to the object to be lifted.

The boom has a 37-foot overhang and is able to put a 16-foot diameter boiler into the middle of a ship 70 feet beam and having from 35 to 40 feet freeboard. The lifting capacity of the derrick is 120 tons, making it the most powerful floating derrick in America.

The back stays are led to carriages traveling on rollers around the base casting. When lifting weights greater than 30 tons, heavy stay-rods hanging from the end of the boom are secured to the edge of the pontoon so that the boom is fixed relatively to it. (These are plainly shown in the annexed cut).

Each member of the boom is a box girder, the upper surfaces of which have sliding ways of planed brass, on which run, on lignum-vitæ slides, the carriage which supports the sheaves. This carriage measures 7½ feet long by 2 feet wide, and supports or bears two lifting-tackles, one a double fall for light lifting and the other a 20-fold purchase with 10 sheaves. The ropes are 1⅛-


ATLAS


Bradley & Porter, Engr's, N.Y.

inch steel-wire cables. The sheaves on the carriage have been so arranged that the leading ropes cannot foul. The heavy hoist is calculated to lift 100 tons, while the lighter hoist will lift about 20 tons.

The light-hoisting-gantlin and carriage-traverse machinery is driven by two 14 x 12 engines coupled. On the engine shaft there are wedge-gear pinions driving into wedge drums, thence to the rope drum. Each movement is controlled separately or together. The main hoisting drum, independent of other movements, is 60 inches in diameter by 8 feet long, and is also driven by two 14 x 12 engines coupled, driving through gearing to drum, which is entirely controlled by these engines.

The reduction of gearing is 1 to 36, having large power in engines, and gives better control of the load so far as speed is concerned. The engines can hoist 100 tons four feet per minute and lower at any desired speed. The general design of all the machinery has been greatly improved from previous ideas by keeping all driving points nearer deck, thus lowering the center of gravity far more than in previous instances. The machinery is placed within the tower on deck. All wearing parts of hoists, carriages, and friction bearings of king post are fitted with interchangeable parts, requiring the least possible effort for alterations or repairs.

THORNYCROFT'S SCREW TURBINE PROPELLER.

[Reprinted from "Engineering."]

We give on the opposite page two illustrations of an improved screw turbine propeller which has recently been introduced by Mr. John Isaac Thornycroft and Mr. Sydney Walker Barnaby. Our readers are acquainted with the original form of screw turbine* devised by Mr. Thornycroft, a form of propeller which has been fitted in its original form to a number of vessels, the best known of which are the steamers built for the Nile expedition. These vessels were 140 feet long by 21 feet wide, and attained the high speed of 17¼ miles per hour on the remarkably low draught of 21 inches.

* See "Engineering," vol. XXXII, page 652; vol. XXXV, page 463; vol. LI, page 246; and vol. LIII, page 731.

The arrangement of the new propeller is well shown by our illustrations, which are two views taken from photographs of a sectional model. The great drawback to the original form was that it would not drive the boat astern in an effective manner. Many devices have been suggested to overcome this difficulty, some of an elaborate and extremely ingenious nature, but it is only lately that the problem has been solved in the manner now illustrated. Referring to our engraving, it will be seen that the propeller comprises a cylinder containing within it a body or boss of such a shape that the channel is gradually contracted from the forward to the after end. Within the forward part of the cylinder, there are revolving screw blades attached to the forward part of the boss. The forward part of the boss is separate from the after part. It will, of course, be understood that the after part is that which ends in a point and projects beyond the cylinder. The ordinary four-bladed propeller (it appears more as a three-bladed propeller in the illustration, being in section) is forward of the screw turbine.

The pitch of the forward edge of the screw blades, multiplied by the number of revolutions, is approximately equal to the velocity of the feed or speed of the screw through the water. The pitch increases uniformly along the length of the blade, thus imparting a uniform acceleration to the water. Aft of the revolving blades are placed numerous fixed blades of contrary curvature. The area of the channel through the propeller is so proportioned as to suit the acceleration of the water caused by the blades. In this way there is secured at the forward end a large opening which will admit a certain quantity of water at the velocity of feed, whilst at the after end the area is restricted to that necessary to allow of the exit of the water at the speed of discharge. The long tapering body forming a prolongation of the boss outside the cylinder is for the purpose of allowing the stream of water to unite gradually without the formation of eddies. As the long pitch of the screw blades causes considerable rotation of the water, the curved guides are so formed as to direct the water into a straight line aft, and the rotary motion is thus utilized without loss. The efficiency of the screw tur-

bine has been found by experiment to be at least equal to that of the ordinary screw, and a given thrust can be obtained with a much less diameter. It is, therefore, a propeller more especially suitable for light-draft vessels.

The water is not accelerated at all before it reaches the propeller; in other words, there is no sucking action like that produced by an ordinary screw, and for this reason there should be less augmentation of hull resistance than with other forms of propeller, and were it not for the large surface exposed to friction, it should have a much higher efficiency than the common screw. It can, as stated, be used with success where shallow draught is required, and might be applied with advantage in sea-going vessels, which are often in light trim and do not then properly immerse their propellers, thus leading to a large waste of power.

The propeller, so far as described, has the objection referred to of not being suitable for going astern. The reason of this will be apparent. The water then enters the annular space at the after end, and issues at the forward end as a revolving ring of water. Moreover, as the channel increases in area towards the forward end, the water which comes in from aft is unable to fill it, so that the middle portion of the cylinder is occupied by dead water, and some loss is probably occasioned by eddying here.

In order to overcome these difficulties, supplementary blades are attached to the shaft at a little distance from the screw turbine proper, their function being to propel when going astern. In order that they should not interfere with the action of the screw turbine when going ahead, the pitch of these blades is so arranged that they simply slice their way through the water without either rotating it or driving it against the screw turbine in rear of them. This effect is obtained by making the pitch multiplied by the revolutions equal to the speed of advance.

When the direction of motion is reversed to the going-astern direction, these blades receive water not only through the turbine, but also from outside it. As the speed of the vessel for a given number of revolutions is always much less when going astern

than when going ahead, the blades throw a stream of water forward and thus propel the vessel astern. If the slip of a screw were constant at all speeds the sternway screw would be inoperative at all speeds ahead, and this is so nearly the case that it is anticipated that, if the pitch be adapted to the speed at which the slip is least, the sternway screw will not retard the vessel appreciably at any speed. In the case of a vessel used for towing varying loads, when the slip will vary as the load, the sternway screw will at times do part of the work of propelling.

Mr. Thornycroft holds the view that the power to go astern is not of first importance in a steam vessel, and he has even proposed to do away with the complication of reversing gear in the marine engine, holding it to be expensive and cumbersome; and although used but seldom, the parts comprising it are at work during the whole time the vessel is under way. That is very true, but reversing gear is very like the Texan's revolver: "You don't want it often, but when you do, you want it badly." Mr. Thornycroft points out that the power of reversing motion is not possessed by even the fleetest animals. Nature provides means for stopping and turning quickly, but not for going astern. Seamen have, however, departed so far from the old sailing-ship days, when it was necessary to check the way of a vessel in good time, and have become so accustomed to facilities afforded by a reversible engine, that they are dissatisfied with a propeller that will not back a ship. It is in indulgence of this weakness in the modern mariner, and to meet other requirements, that the sternway screw has been devised. We may say we so far agree with Mr. Thornycroft and Mr. Barnaby that we believe there would be fewer accidents if marine engines were not reversible; but this would be because fewer things would be attempted. For instance, the Thames 'tween-bridges passenger traffic, the New York ferry services and many other services of this nature would be impossible.

The sternway screw was tried on the *Goodwill*, a vessel built for the Baptist Missionary Society, and is said to have answered its purpose admirably.

MANGANESE STEEL.

In a lecture delivered before the Franklin Institute, February 20, 1893, Mr. H. M. Howe described the properties of manganese steel. Among the points brought out by the lecturer was the fact that the strength and ductility of the material were both increased by hastening the cooling of the metal. In doing this in the case of manganese steel castings, however, it is necessary to take care that the rapid cooling does not cause cracks at re-entrant angles and other weak spots. In forging ingots of the steel the temperature, Mr. Howe says, should be raised gradually, as the metal conducts heat slowly. During forging, the metal acts like a carbon steel containing 1.25 to 1.50 per cent. of carbon, and requires heavier blows than most steels. It may be rolled hot into sheets .049 inch thick without special difficulty, and with proper precautions to as thin as .014 inch thick. Cold, it has been rolled still thinner, but requires frequent annealing. In machining it the hardest carbon or chrome steel should be used, Mushet and other self-hardening steels being suitable. Light cuts and a very slow feed are essential. As an estimate, it takes four times as long to machine manganese steel as common carbon steel. At present the metal has been very successfully used for the pins of dredger buckets, as it resists abrasion in a remarkable manner, wearing only about one-sixth to one-eighth as fast as the best carbon steel. For armor plating, preliminary trials with thin plates show that to offer the same resistance as a manganese steel plate, a wrought-iron plate would have to be 75 per cent. thicker. The metal has also been used for car wheels; 22 such have run on an average 168,653 miles before the first turning. These results were obtained on a New England line. This figure, Mr. Howe says, is seven times the average mileage of chilled cast-iron wheels on this line, which is a peculiarly trying one owing to the frequent stops.

THE STRENGTH OF TORPEDO BOATS.

["Engineering," January 20, 1893.]

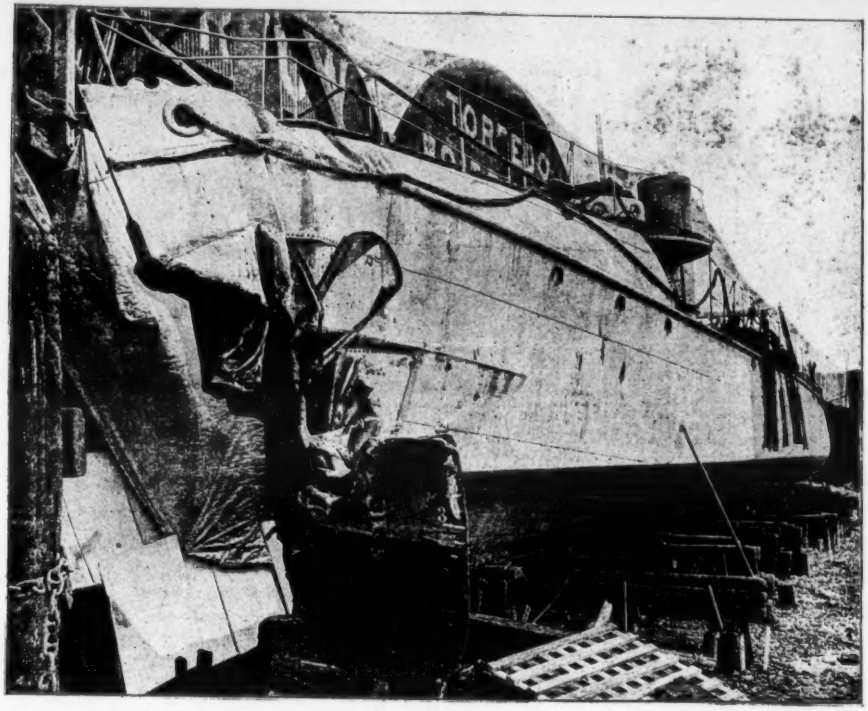
The annexed illustration prepared from a photograph shows the bow of a first-class torpedo boat which had been in collision

with a sailing barge. The matter is interesting as affording a record of the amount of rough usage these lightly-built craft will stand without absolute destruction. The vessel in question was built by Messrs. Yarrow & Co. for the Government of Victoria, and at the time of the accident was on her trial trip. She is 130 feet long by 13 feet 6 inches wide, and had a displacement at the time of the trial of about 80 tons. The hull is divided into ten water-tight compartments, the bulkheads forming the divisions being of steel and carefully fitted; and as, in addition, they are pierced by neither doorways nor sluices, the compartments are in truth water-tight. It was this fact, coupled with the excellent wormanship of the structure, which doubtless saved the boat from sinking when the accident occurred.

The machinery of the boat consisted of three-stage compound engines capable of giving out about 1,100 indicated horse power. There was one large locomotive boiler producing steam at about 170-pounds pressure. The material of which the hull is constructed is mild steel. It has a tensile strength of about 27 tons, combined with great ductility. The latter is a point to which Messrs. Yarrow have always paid great attention; that it was so in the case of the boat in question a glance at the engraving will amply prove.

A description of the accident by one who was on board at the time says:

"We left the builders' yard at 11 A. M., and ran down to Gravesend at quite an easy speed for a torpedo boat, viz., about 17 knots. At full speed, the swell made by torpedo boats is so great that the high speed trials are not permitted to be made above Gravesend, and it was only after we had passed that place that we proceeded to raise steam to full speed. By 12·30 we were at full speed, and went up and down the measured sea-mile in the Lower Hope in fine style, the mean of about 12 runs giving us a speed through the water of rather above 23 knots, with between 160 pounds and 170 pounds of steam, and rather over 400 revolutions. The vibration, even right aft, was nothing more than often experienced in passenger steamers, a fact due to the system of balancing the reciprocating parts of the engines


TORPEDO

introduced by this firm. We had been at full speed for about two and a half hours, and had been to the Nore and back again to the Mucking Lighthouse, when, turning round to go down Sea Reach, something went wrong with the steering gear. A clutch had worked back without its being noticed, and the consequence was the steersman lost all control, and, before the engines could be got to fairly astern, we ran, stem on, at about 18-knot speed, into a large wooden sailing barge, loaded with over 100 tons of wheat, at anchor just above the Lower Hope Point beacon.

Our stem cut deep into the forepart of the barge, just forward of the mast. The bargees got into their boat, and our engines were put full speed astern to disengage us, for the anchor of the barge held her so firmly that we could not push her on the shore, although this was only 80 yards distant. The sinking barge was so firmly fixed to our stem that she was pulling our bow into the water and lifting our stern so much that our propeller was only skimming the water, and we were helplessly gripped by the nose like a bull by a bulldog. To make matters worse, the strong ebb tide acting on our side, twisted our stem completely round to port, at right angles, tearing our plates open as far as the second bulkhead. For a short time, it seemed as if nothing could save us, when the barge heeled suddenly right over to port, tore herself clear of the wreckage of our bows, and sunk in some 20 feet of water, still held by her anchor.

"As soon as we were released from the weight of the barge, our bows came up and our propeller gripped the water. We then steamed ahead to clear some vessels at anchor below us. We now found that the third bulkhead from forward was quite watertight, and by trimming all our ballast right aft we got the wreckage of our stem sufficiently out of the water to enable us to secure it by means of chains, &c.; for, until we had done this, it would not have been safe to steam ahead; indeed, our cutwater, being all twisted over to port, acted like a bow rudder and prevented us steering properly. An hour's work in hauling up the torn plates of our bow enabled us once more to obtain steering control. Engines, boilers and steam pipes were not at all injured, and we steamed home without aid at the rate of quite ten

or twelve knots per hour, arriving late, but all safe, at the works at Poplar."

It might have been expected that such a collision would have shifted the machinery and broken some of the steam joints, but such, it will be seen, was not the case. On account of the lightness of the scantling of the bow, no solid resistance was presented to cause the stoppage of the boat, the momentum being gradually taken up, as it were, by a buffer. The scantling of the hull was of the strength customary in Messrs. Yarrow's boats, and, although this may appear exceedingly light, it has been found sufficient in this case to withstand the stress of a voyage to Australia, the vessel in question arriving there some months ago without a rivet leaking.

EFFECT OF SHELL FIRE ON TANKS OF PETROLEUM RESIDUUM.

"Le Yacht" of February 25th states that experiments have been made at Toulon to determine the safety of tanks of liquid fuel when exposed to shell fire. 64 shells from a 47 mm. R.F.G. were fired into a tank filled with astatké or petroleum refuse, such as is used for fuel, without any harm resulting.

It appears, therefore, that the powder charge of one of these small shells will not ignite the liquid fuel. It would seem not improbable that the same might hold good for larger shells, but further experiments are to be made to determine this point accurately.

NEW SHIPBUILDING PROGRAMME FOR BRITISH NAVY.

[Extracts from comments in "Engineering" on the proposals of Earl Spencer, First Lord of the Admiralty.]

To return, however, to Lord Spencer's proposals. We find that the two new battle-ships are to be of the type of the *Royal Sovereign*, "with certain improvements." There will be four 12-inch breechloading new type guns, "mounted in a manner that will give substantial advantages over any previous mountings." That mounting will be looked forward to with great interest, for there is doubtless much required in the method of

mounting of heavy guns, although many of the details connected with them are very beautifully carried out. In secondary armament, these two ships will be superior to that of all existing battle-ships in both power and protection. That is promising a good deal, and as secondary armament has advanced enormously within the last two or three years, we can easily credit the statement, knowing something of the work that has been done in this direction of late. The second-class cruisers are to be improvements on the *Astræa* class. They are to be more powerfully armed than the latter class of vessel, and, in addition, are to be better protected, and are to possess large coal endurance. This is all very pleasant, but these descriptions always read so well in the statement. The two sloops are to be improvements on the gunboats of the *Goldfinch* class, on which, we are told, favorable reports have been received. These sloops are much wanted to take the place of older vessels on foreign stations.

The two cruisers to be let out to contract are not described further than to say that they are "to embody the results of experience gained with our existing cruisers, particularly of the *Blake* and *Edgar* class." We are, however, assured by the statement that "in speed, coal supply, armament, and defence they are to surpass any cruisers built or building."

The *Renown*, which is the new battle-ship lately commenced at Pembroke under the directions of the late Board, is briefly referred to in Lord Spencer's statement. She is described as an improved *Centurion*. Her length is to be 380 feet and extreme breadth 72 feet. The displacement will be 12,350 tons. With natural draft, on the eight-hours' contractors' trial, a mean speed of 17 knots is anticipated. With moderate forced draft, a maximum speed of about 18 knots will be reached. This vessel's armament will consist principally of four 10-inch breech-loading guns carried in two barbettes and mounted similarly to those of the *Centurion*—that is to say, in barbettes. Her secondary armament is described as "extremely powerful," including ten 6-inch quick-firing guns. In the protection of the secondary armament, the arrangements of the *Renown* are more efficient than those in any

preceding battle-ship. The armament of submerged torpedoes will also be more extensive than in any other battle-ship. The barbette armor will be somewhat thicker than that in the *Centurion*. The hull armor will be arranged on an entirely different principle from that which has been adopted in the *Centurion*, and associated with a different arrangement of the protective deck. This point has received the most careful attention; and it is considered by the authorities that the disposition and thickness of hull armor adopted, together with the improved qualities of armor now obtainable, will make the ship well adapted for close action. Like the *Centurion* and the *Barfleur*, the *Renown* will be wood-sheathed and coppered, and will be capable of passing through the Suez Canal.

The 14 torpedo vessels which are proposed will not be ordered just yet. The estimates of the present year included a provision for 10 first-class torpedo boats, and it was proposed to spend upon them £20,000. A large scheme, we are now told, had been previously under consideration, the carrying out of which was dependent upon financial considerations. Early in the financial year, the Admiralty decided, with the concurrence of the treasury, to begin the construction of these vessels, and to accelerate the progress of the first-class boats. Orders for the latter were therefore given to certain firms, and these boats are now fairly forward owing to the exceptional activity with which the contractors have taken up the work. In speed and power of keeping the sea, they will be superior to all their predecessors —at least, so we are informed by the statement, which evidently pretends to some prescience in these matters.

In addition to the first-class boats, 6 torpedo-boat destroyers of new type have been ordered with a guaranteed speed of 27 knots per hour. The orders have been placed with three private firms of high reputation, and it is anticipated that in speed, armament, and sea-going capability these vessels will be superior to any yet built. It is proposed to complete and try the first examples of the new type next summer, and subsequently to order 14 other vessels of the class from private firms during 1893–'94.

With regard to reconstruction and repairs, we are told that the *Téméraire*, *Ajax*, *Bellerophon*, *Osborne*, *Kingfisher* and *Bramble* will be completed in 1892–'93 at the home yards; the repair of the *Agincourt* and *Northumberland* has been commenced. The reconstruction of the *Rupert* and *Hercules* will be completed in 1892–'93, and that of the *Devastation* in 1893–'94. The *Monarch* will be well advanced in 1893–'94, so as to admit of her completion early in 1894–'95. The *Sultan* has also been taken in hand during the present year, and will be completed in 1894–'95. The bringing of these antiquated but perfectly sound vessels into line again is one of the best steps taken of late at the Admiralty. There is, we believe, a saying that, if one keeps anything for 21 years, a use is sure to arise for it during that period. The law seems to apply to war vessels, the changes in armament bringing back to use types and arrangements that have become obsolete.

In that part of the report headed "Naval Ordnance and Torpedoes," the chief passage of interest is the statement that a 12-inch breechloading steel and wire gun of a new design, intended for the armament of the new battle ships, is now under manufacture. It is said in the statement to be relatively, as regards weight, a much more powerful weapon than any previously constructed in this country. It will be provided with an hydraulic mounting, capable of being worked entirely by hand power in case of accident or failure of any part of the mechanism, and with an all-round loading position. The ability to work by hand is, of course, a desirable feature if it can be obtained without too great a sacrifice, but the demand for hand working may easily be pushed too far. It must be remembered that a man (still more a number of men) does not provide an indestructible motor for gun working.

In regard to "Boilers and Machinery," which supply a subhead in the "statement," there is a more jubilant note than has been heard of late. This is, of course, due to the "Chatham ferrule," a simple device which has brought more peace of mind than many a grander and more elaborate mechanical invention. The statement notes with satisfaction that the failure of some double-ended boilers of the common-combustion-chamber type

has not led, as was at one time thought likely, to the necessity of ordering new boilers, at a very heavy cost, for the *Devastation*, *Thunderer* and *Vulcan*. Experiments, we are told, happily resulted in the discovery of the simple device of a capped ferrule, or short tube, placed inside the mouths of the boiler tubes, which enables the heat generated to be more effectually distributed through the tubes, and so prevents local overheating.

The reference is, of course, made to the now well-known ferrule which was evolved at Chatham, and was, as stated in the report, first fitted to the ill-fated *Barracouta*. The result, as we have already stated, was that the full power was realized, the statement being made that they could not make the tubes leak if they tried. From an Admiralty point of view, the ferrule has been a stupendous success. It has bridged a difficulty which was of a very serious nature. That it does not satisfy orthodox—perhaps "old-fashioned"—engineers is doubtless a small matter, because the ships can now run their trials, and what more can be asked? The ferrules, we are told, were also "fitted to the *Thunderer*, which vessel, after going through her natural and forced-draft trials most satisfactorily, steamed to Madeira and back at about four-fifths natural-draft power. The boilers of the *Devastation* and *Vulcan*, and all boilers of this type are being so fitted. These ferrules may be found useful also in the case of boilers of other types, as it is expected that they will prove a valuable adjunct in preventing the leakage of tubes resulting from the accidental or other accumulation of grease or other deposit on the heating surfaces of the boilers."

Reference is made also under the same heading to the steps that have been taken to introduce experimentally a water-tube boiler into the Navy. It is a matter in which this country is a long way behind; but something—a little—at last is being done. Messrs. Thornycroft and Co. are building the *Speedy*, a torpedo gunboat, which is to have the Thornycroft boiler. This will be an example on a fairly big scale. Messrs. Yarrow and Co. are also introducing water-tube boilers in some of the craft they are constructing, so that the torpedo-boat builders, as is usual, are initiating another new departure in marine practice. Messrs.

Laird have also included water-tube boilers in their contract with the admiralty, but they have gone abroad for their design. We think that British engineers, as a rule, have not very full information on what has been done abroad in this branch of engineering. Some of the big French ocean-going ships have introduced the system, apparaently with good results, certainly in the matter of speed.

SHIPS.

UNITED STATES.

Indiana.—The battle ship *Indiana* was launched from the yard of the William Cramp & Sons' Ship and Engine Building Company, at Philadelphia, on Tuesday, February 28.

The *Indiana*, the first of the three coast-line battle ships to be launched, was authorized by act of Congress approved June 30, 1890, appropriating for the construction of three battleships at a cost not exceeding $4,000,000 each. Bids were opened October 1, 1890, and on November 19, 1890, contracts were awarded to the William Cramp & Sons' Ship and Engine Building Company, of Philadelphia, for two of them, at a contract price of $3,020,000 each, to be finished within three years.

The vessel is of great fighting power united with adequate protection, able to take the sea in all weathers, and with draught suited to work in our shallow harbors.

The hull is protected by belts of heavy armor 7½ feet wide, turned in forward and aft to sweep round the bases of the armored redoubts, the whole including about 75 per cent. of the water plane area. Forward and abaft this belt are heavy underwater protective decks, sloping at the sides to 4½ feet below the water. There is also an armored deck over the belt. Coal bunkers are worked over the belt deck, and belts of water-excluding material are worked on the slopes of the armor deck forward and aft. Above the belt armor, and extending from redoubt to redoubt, the side is protected by five inches of steel.

The vessel is cut up forward, making a powerful ram bow, and doing away with excessive bow waves, on account of the easier lines so obtained, as well as greatly adding to the maneuvering power.

Length on the water line,	348 feet.
Breadth, extreme,	69¼ feet.
Draft, forward and aft,	24 feet.
Displacement,	10,200 tons.
Maximum speed,	16.2 knots.
Sustained sea speed,	15 knots.
Displacement per inch (tons),	42.

Scantling and general construction.—The vertical keel is 20 pounds with 4 by 4 inch angles of 13-pounds on the lower edge, and 3 by 3½ inch angles of 8-pounds on the upper edge. The flat keel plates are: outer, 30 pounds; inner, 25 pounds. The stem, stern posts, struts and rudder frame will be made of cast steel or the best hammered scrap iron.

The transverse frames are spaced 4 feet apart in the double bottom and 3½ feet at the ends. From armor shelf to keel the frame angles are 5 by 3½ inches of 12 pounds per foot; the reverse angles 4½ by 3 inches of 10 pounds per foot, with bracket plates of 15 and 13 pounds and lightened plates of 12½ pounds. The outer angle is continuous, the reverse frame angle being in short lengths between the longitudinals. The water-tight floors are of 12½ pounds, with 3½ by 3 inches of 8-pound box angles.

Before and abaft the double bottom, and below the protective deck, the frames are made of channel or Z-bars 6 by 3½ by 3½ inches, of 15 pounds per foot, with 12-pound bracket plates. Above the side armor and the protective deck at the ends, the frames are 6 by 3½ by 3½ inches of 15 pounds, with alternate frames of the same size in wake of the casemate armor.

The longitudinals are generally of 17½-pound plates, with 8-pound angles on the outer edges and 7-pound angles on the inner edges; the third longitudinal, which is water tight, however, is of 20 pounds, and the shelf plate of 25 pounds.

The main deck beams are of 10 by 3½-inch angle bulbs of 27 pounds per foot; the berth deck beams of 9 by 3½ inches of 22-pound angle bulbs, increased to 24 pounds where coming under armor; the protective deck beams are 9 by 3½ inches of 24 pounds per foot; the platform beams are of 10-pound angles or

11-pound channel bars; the superstructure deck beams are 9 by 4 inches of 23½ pound bulbs; the bridge beams are 9 by 3½ inches of 21.8-pound angle bulbs.

The bottom plating for a length of 230 feet amidships is 25 pounds per square foot up to the water line; forward and abaft these limits it is 23 pounds. Above the water line it is 18½ pounds throughout, thickened wherever necessary on account of the chafe of the anchors, blast of guns, etc. The flat keelson plate is 15 pounds; the inner bottom, as far as third longitudinal, 14 pounds per square foot, thence to armor shelf 10 and 7½ pounds per square foot.

There is a middle-line bulkhead of 12 and 10 pounds per foot, well stiffened horizontally and vertically, between the two engine rooms and the fire room on each side. The vessel is divided into a great number of water-tight compartments by means of longitudinal and transverse bulkheads of 10 and 12 pound plates.

The decks are all plated over with plates not less than 12½ pounds to the foot, and the main and superstructure decks are covered with 3-inch deck planks.

Armor.—There is a water-line belt 3 feet above and 4½ feet below the water line of 18-inch armor. Rising from this at each end are armored redoubts of 17 inches in thickness extending above the main deck 3½ feet, giving an armored freeboard of 15 feet 2 inches. These redoubts protect the turning gear of the turrets and all the operations of loading. The turrets are inclined, 17 inches thick, powerfully strengthened. The horizontal thickness of the turrets is 20 inches. The side armor belt is backed by 6 inches of wood, two ¾-inch plates, and a 10-foot belt of coal. The vessel above the belt has 5 inches of armor protection.

The 8-inch guns have barbettes of 10 inches, inclined turrets of 8½ inches, and cone bases and loading tubes of 3 inches.

The 6-inch guns are protected by 5 inches of armor and have 2-inch splinter bulkheads worked around the deck, inside of which the ammunition for these guns is set up.

The 6-pounders, where mounted between decks, have 2-inch armor worked around them; elsewhere they have the usual ser-

vice shields. The 1-pounders are protected by two inches of steel.

The deck over the belt is $2\frac{3}{4}$ inches, and at the ends the underwater decks are 3 inches. These decks are made up of two thicknesses of $\frac{1}{2}$-inch mild steel plates, with the remaining thickness all in one plate of steel armor.

There is a 10-inch armored conning tower, with a 7-inch tube for protecting the voice pipes, electric wires and steering connections.

Armament.—Four 13-inch 35-caliber breech-loading rifles; eight 8 inch breech-loading rifles; four 6-inch breech-loading rifles; twenty 6-pounder rapid-fire guns; six 1-pounder rapid-fire guns; two Gatlings, and six torpedo tubes or guns.

This represents a weight of armament superior to that of any of the latest battleships laid down by foreign powers.

The 13-inch guns are $17\frac{3}{8}$ feet above the water, and have great arcs of train. The 6-inch guns are 14 feet 10 inches above the water, and all fire across the center line. The 8-inch guns are mounted $24\frac{3}{4}$ feet above the water, and can fire over the tops of 13-inch guns, having a train of 14° across the middle line These guns can pierce at two miles the armor of many of the modern armored cruisers of great coal endurance. They can be brought into action early in the engagement on account of their great height, and can be used with great effect against the lighter armored and unarmored parts of the heavier battleships.

Special attention has been paid to the ammunition supply, and a rapid, efficient and thoroughly protected supply has been secured, and one which it is believed is an advance upon all systems now in vogue.

The powerful secondary battery is protected in part by shields built in the vessel and in part by shields on the mounts, and is so disposed that a stream of projectiles will radiate from the vessel, which will lead to the almost certain destruction of any light boat venturing within range.

Fixed torpedo tubes are carried at the bow and stern, and two training tubes firing through 5 inches of protection are carried on each broadside.

All modern appliances for offensive and defensive purposes, such as torpedo defence nets, search lights, etc., are carried, and weight is reserved for any new fittings or devices which the constant progress in the art of war will most certainly develop as the work progresses.

Great care has been taken in disposing this great battery that one gun will not interfere with the line of fire of another; the boats have been stowed amidships where the blasts cannot reach them, and special strengthening has been given to the sides and deck, that they may withstand the effect of the great strains brought upon them by the fire of the larger guns. The 13-inch guns are kept 6½ feet above the deck at the middle line.

An armored conning tower of 10-inch plates is built forward, having an armored tube to protect the communications, by means of which the commanding officer will direct the movements of the vessel, guns and men. Rising from the conning tower is a military mast, carrying two tops for rapid-fire and machine guns, the ammunition being sent to them inside the mast.

Lighting, drainage and ventilation.—An electric plant will be placed on board to furnish all necessary electric energy for lighting the vessel, working the search lights, and supplying electric motors.

The drainage system is so arranged that any compartment can be pumped out by hand or steam, and the system of drainage has been much simplified. Powerful wrecking pumps are carried in the donkey boiler rooms above the protective deck so that, in case water enters the fire rooms below, steam can still be utilized in pumping out the vessel.

A thorough system of ventilation is installed, and air can be forced into or drawn out of any part of the vessel by means of steam fans of great capacity, while special means are to be provided for such places as the donkey boiler, dynamo and hydraulic pumping rooms, and for removing the gases from coal bunkers.

Quarters for officers and crew.—The captain has one large cabin and two state rooms, with adjoining bath rooms, so that in case an admiral may wish to transfer his flag to these vessels he will have comfortable quarters.

There are 16 ward-room state rooms on the berth deck opening into a passage which leads to the ward-room mess room. This mess room is lighted from above, and is against the side of the vessel, with a number of air ports opening into it. Adjoining it is a large pantry furnished with all the necessary sinks and fittings, and a steam table for keeping food warm after it is brought from the galley. The ward-room officers have a lavatory fitted with tubs and shower baths. The junior officers have a large compartment similar to the ward-room mess room, and the torpedo room adjoining will be made as a country. They have a bath room similar to that of the ward room. On the starboard side of the berth deck just forward of the countries are the water closets for all the officers. On the forward berth deck is a large berthing space for the crew, left open as one large room. In this room alone there are about 100 cubic feet for every man in the crew, while in other parts of the berth deck and under the superstructure a large number of men can be berthed. In the forward end of this berthing space are a number of wash-basins and shower baths, and just forward of them, separated by a water-tight bulkhead, are the water closets for the men.

Offices, store rooms, etc.—The usual offices are fitted and a large chart house has been built containing a berth, wash-bowl, transoms, tables, etc., in addition to the steering wheel, telegraphs and voice pipes. The store-rooms are ample for stowing provisions for a crew greatly exceeding the one contemplated for three or four months.

The coal bunkers will stow 1,800 tons of coal, sufficient to carry the vessel at 10 knots 16,000 miles.

Machinery.—The propelling machinery is capable of developing, when forced, 9,000 I.H.P., and under ordinary conditions 8,000. There are two propellers.

The main engines are inverted, vertical, direct-acting, triple-expansion, with cylinders of 34½, 48, 75 inches in diameter and 42 inches stroke, and it is estimated that at a piston speed of 9,000 feet per minute, or 129 revolutions, the I.H.P. will be 9,000.

Each engine and its auxiliaries is separated from the other by a fore-and-aft water-tight bulkhead, so that in case of accident to one engine the other would not be affected.

The main valves are of the piston type, worked by Stephenson, double-bar links; one valve for each high-pressure cylinder, two for each intermediate, and four for each low-pressure cylinder, the diameter of all the valves being 17 inches.

The intermediate and low-pressure cylinders are steam jacketed at sides and bottom, the high-pressure ones have linings, but are not jacketed. The framing consists of an inverted Y-column at the back of each cylinder and two forged steel cylindrical columns at the front. The engine-bed plates are of cast steel supported on wrought steel keelson plates which are built in the ship.

The piston rods, valve stems and all working rods are of mild forged steel; the pistons are of cast steel.

The crank shafts are of mild forged steel; they are in three sections, which are reversible and interchangeable with each other and with those of the other engine. They are 14 inches in diameter in the journals and 15 inches in the pins, and have axial holes of 6 inches in diameter through the former and of 6½ inches through the latter.

There is a separate condenser for each engine, made with cast brass heads and shell. The heads are seven-sixteenths inch in thickness and will contain all the nozzles for steam and water openings. The diameter is 5 feet 9 inches, and length between tube sheets, 10 feet 3 inches. Each condenser contains 3,788 seamless drawn brass tubes, five-eighths inch outside diameter, giving 6,353 square feet of cooling surface on the outside.

Condensing water is supplied each condenser by a centrifugal circulating pump capable of discharging 9,000 gallons of water per minute from sea or bilge.

The air pumps are of the Blake vertical type, with two steam cylinders 12 inches in diameter, and two water cylinders 25 inches in diameter, the stroke being 18 inches. The steam and water pistons on each side are on the same rod, and the two rods are connected by links to a vibrating beam, pivoted in the center. The valve gear of the steam cylinders is driven from a pin on the beam near the center. These pumps are similar to those on the *New York*, which on the preliminary trials have given great satisfaction.

There are four main and two auxiliary boilers, made of mild steel, of the horizontal, return, fire-tube type, and intended for a working pressure of 160 pounds.

The main boilers are 15 feet outside diameter and 18 feet long, double-ended and with shells $1\frac{19}{64}$ inches in thickness. Each boiler contains eight corrugated steel furnaces 3 feet internal diameter. The tubes are lap-welded steel, $2\frac{1}{2}$ inches outside diameter; the ordinary ones No. 12, B.W.G., and the stay tubes No. 6, B.W.G., in thickness. The main boilers have about 17,240 square feet of heating surface and about 552 square feet of grate; the auxiliary boilers are the same as those designed for the *New York* and contain 1,953 square feet of heating surface and 64 square feet of grate. The main boilers are arranged in groups of two, with two smoke-pipes. The auxiliary boilers are above the protective deck.

The ship is intended to cruise under very light forced or natural draft, and has been given large heating surface for this reason.

There is a main and an auxiliary feed pump in each fire room with an additional auxiliary feed pump in each engine room, the whole capable of supplying many times the quantity of water needed at full power.

There are evaporators, distillers, auxiliary pumps, blowers for draft and ventilation, and the other auxiliaries now furnished ships of this class.

The propellers are of manganese bronze, with adjustable blades.

New York.—The official trial of this vessel is expected to take place the latter part of this month. It will consist of two runs in opposite directions, over a 40-knot course laid out between Cape Ann and Cape Porpoise (on the Maine coast), where there is water of ample depth. A number of vessels will be stationed along the course to take current observations.

The vessel was taken to sea by the contractors for a set of preliminary trials lasting from March 21st to 25th, during which the working of the machinery was entirely satisfactory.

The boilers were not forced to the limit at any time, as the maximum air pressure carried did not exceed $\frac{7}{8}$-inch, while the contract allows $2\frac{1}{2}$ inches. Several runs were made between

lighthouses to get an idea of the speed. The mean of two runs over a course 9.88 knots long gave 20.12 knots. As there was no corps of observers, it is, of course, impossible to give accurate data of steam, etc., but it may be mentioned that the revolutions were counted twice and gave 117 and 124. When making 117 revolutions the steam was 140 pounds; 1st receiver, 47 absolute; 2d receiver, 19 absolute; with 124 revolutions the steam was 160, but the H.P. links were run in and the throttle was not wide.

There is every reason to believe that the vessel will make in the neighborhood of 21 knots on the official trial in deep water.

It is interesting to note that, from the experience gained on these trials, it was decided to lengthen the smoke pipes, both for increasing the draft and to carry the smoke clear of the military tops. They are now over 90 feet high or very little less than those of the *Brooklyn*.

Detroit.—This vessel was tried over a course nearly 36 knots long in Long Island Sound, April 17th, making two runs, one in each direction. Tidal observations were made at five points, dividing the course into approximately by equal divisions.

The first run was made against the tide, whose total force is reported as 0.952 knots, in 1 hour 58 minutes and 22.87 seconds. The second run, with the tide, was made in 1 hour 48 minutes 50.5 seconds, the tide being reported at 2.057 knots. The mean speed was consequently reported at 18.71 knots.

Owing to the fact that the speed premiums are paid only for completed quarter knots, the contractors will receive the premium for only 18.5 knots, or, in this case, 1.5 knots, amounting to $150,000.

The averages of the machinery data for the whole trial are:

	Starboard.	*Port.*
Steam at boilers, per gauge	180.7	
Steam at engines, per gauge	164.0	162.7
Steam at first receiver, absolute	80.9	76.7
Steam at second receiver, absolute	26.8	27.9
Vacuum, in inches	25.1	25.4
Revolutions per minute	170.1	170.1
I.H.P. main engines	2,599.48	2,554.79
I.H.P. main engines and auxiliaries	5,227.14	

Prior to the runs over the measured course, a series of progressive trials was conducted over the standard mile at Newport. These runs were for the information of the Department, and were not used in the final trials, although the revolutions for the two runs were carefully observed. Although the curve of speed and revolutions has not yet been carefully laid out, it may be said that the speed, as determined by the average revolutions, agrees very closely with that determined by the runs over the long course.

Owing to lack of time for preparing the data, the full account of the *Detroit's* trial will not be given until the August number, when an article will appear giving the trials of this vessel and her sister, the *Montgomery*.

ARGENTINA.

9 de Julio.—The steam trials of the *9 de Julio* were made off the coast north of the River Tyne in February, 1893. The vessel has been constructed by the firm of Sir W. G. Armstrong, Mitchell & Co. from designs by their naval architect, Mr. Philip Watts, and in this vessel the firm has again broken the record for high speed in warships.

This firm may fairly claim to have initiated this class of vessel in the high-speed cruiser *Esmeralda*, and they have maintained the lead they then took down to the present time. Of the last three cruisers of high speed constructed at Elswick, viz., the *Piemonte*, *25 de Mayo* and the *9 de Julio*, whose steam trials have just now been made, each has in succession attained a greater speed than its predecessor, and at the present time these three vessels are the fastest warships afloat.

The vessel is in the main similar to the Argentine cruiser, *25 de Mayo*, which left the Tyne about the middle of 1891. She is 350 feet long between the perpendiculars, 44 feet broad, 16½ feet mean draught, and her load displacement is 3,500 tons. These dimensions are repeated in the table given below with the dimensions of the *25 de Mayo* and other vessels, so that they may be readily compared with one another.

Her contract powers with natural and forced draft were 9,000 indicated horse-power and 14,500 indicated horse-power, respectively; and, as will be seen below, the former of these was more than realized, and the latter might have been considerably exceeded if it had been desired to do so. The contract speeds with natural and forced drafts were 21½ knots and 22½ knots, respectively, and these also have been much exceeded on the trials.

Her armament consists of four 6-in. quick-firing guns, three of which fire right ahead and one right astern; eight 4.7-in. quick-firing guns, 12 3-pounder Hotchkiss guns, and 12 1-pounder Hotchkiss guns. She is also provided with five torpedo tubes for 18-in. Whitehead torpedoes. Her propelling machinery, which has been manufactured by Messrs. Humphrys, Tennant and Co., consists of two sets of vertical triple-expansion four-cylinder engines driving twin screws, and she has eight single-ended boilers. Each set of engines has two low-pressure cylinders, each 66 in. in diameter; one intermediate cylinder 60 in. in diameter, and one high-pressure cylinder 40 in. in diameter, the length of stroke being 30 in. in each case. The whole of her machinery and magazines are below the water line and are protected by a strong armor deck, which extends the whole length of the ship. This deck has sloping sides of steel 4½ in. and 3½ in. thick, while the horizontal parts are 1¾ in. thick. She is also provided with a raft body extending her whole length to a height of 4 ft. above the water line amidships, which may be packed with water-excluding material, thus insuring to the ship buoyancy and stability when riddled in action.

She carries 350 tons of coal as her normal supply, *i. e.*, the quantity of coal corresponding to the load displacement above stated, but the bunkers are capable of carrying nearly 800 tons, which would give the vessel a radius of action probably exceeding 10,000 knots at the most economical speed.

The steam trials included a continuous run of six hours' duration with open stokeholds, and, in order to secure daylight for this trial, the count was commenced some time before the engines had attained their maximum power. Notwithstanding this, dur-

ing the six hours the port engines indicated 53,827 revolutions and the starboard engines 52,932 revolutions. The speed corresponding to this number of revolutions, obtained from a curve of revolutions and speed, was 21 94 knots. In the course of this trial a number of runs upon the measured mile took place, the mean revolutions per minute during these runs being 149.2, and the mean speed, taken in accordance with the British Admiralty practice, was 22.028 knots. The mean indicated horse-power developed during the six hours' trial reached approximately 10,300 horses. The curve of revolutions and speed above referred to had been obtained by a series of progressive trials previously made. The vessel was run on the measured mile a number of times at various speeds, and a diagram was constructed by plotting the mean revolutions corresponding to each series of runs upon abscissæ representing the corresponding mean speed, and a curve was drawn through the points thus obtained. It was thus demonstrated that the vessel had maintained a speed of practically 22 knots over the six-hours' run.

As in the case of the *25 de Mayo*, the Argentine Commission attached very little importance to the forced-draft trial, and no endeavor was made to ascertain the utmost speed that could be attained by the vessel under such conditions. At the close of one of the days' steaming, however, with very little additional pressure (about one-half of what would be permitted by our own Admiralty) several runs on the measured mile were made, when a power of about 14,500 indicated horse-power was developed, and notwithstanding that the ship was run in somewhat shallow water, a mean speed of 22.74 knots was attained. There can be no doubt that, under Admiralty rules and conditions, a speed considerably exceeding 23 knots per hour might have been reached.

The trials of the ship included the firing of the guns to test the strength of the structure of the hull and fittings. In the first place three rounds were fired from each gun, one with extreme forward training, horizontally, one on the beam with extreme depression, and one with extreme training aft, and with extreme elevation. Broadsides were also fired from each side,

and the 6-inch guns mounted on the center line on the forecastle and poop were fired horizontally right forward and right aft, respectively. Severe as these trials were, no damage was done to the vessel or fittings.

The guns and their mountings, which are of the most recent Elswick pattern, worked perfectly, and no hitch of any kind occurred throughout the trial.

For purposes of comparison we give the following table of particulars of the performances of similar cruisers of high speed.

Navy.	Ship.	Displacement in tons	Length in Feet.	Speed knots.		I.H.P.	
				Forced draft.	Natural draft.	Forced draft.	Natural draft.
Argentine ...	*25 de Mayo*.....	3,200	325	22.43	21.237	13,800	8.700
Argentine ...	*9 de Julio*.....	3.500	350	22.74	22.028	14.500	10.000
British.....	*Medusa*.	2,800	265	19.9	18.005	10,000	6,300
British.....	*Blenheim*.....	9 000	375	21 6	20 4	21,400	14,900
British.....	*Edgar*	7,350	360	20.97	19.25	12,550	10 200
British.....	*Latona*	3.330	300	20.5	19.35	9 600	7,400
French.....	*Surcouf*	1,850	312	20 51	17.3	6,287	3.508
French.....	*Forbin*.....	1,850	312	20.64	.....	.....	.....
French	*Cecille*.....	5.670	380	19.436	.....	10,680	.....
Italian	*Piemonte*.....	2,500	300	22.3	20.4	12,700	7,000
Spanish.....	*Reina Regenta*.	4,800	317	20.6	18 68	11,500	.....

—[Reprinted from "Engineering."]

ENGLAND.

Grafton.—The Naval Defence Act authorized the construction, in addition to numerous other vessels, of nine first-class protected cruisers of upwards of 7,000 tons displacement. These nine cruisers, the last of which was launched in September, 1892, are of two types. One type, which, being sheathed and coppered, is slightly heavier than the other, displaces 7,700 tons, and is represented by the *Crescent*, *Gibraltar*, *Royal Arthur* and *St. George*. The other type displaces 7,350 tons, and is represented by the *Edgar*, *Endymion*, *Hawke*, *Thescus* and *Grafton*. The last named of these was launched on January 30, 1891, at the yard of the

Thames Ironworks Co., Blackwall. It is usual for contract-built vessels either to be completed for sea at one of the Royal dockyards, or, after having been completed up to a certain point by the builders, to be taken to a Royal dockyard and there finished in accordance with naval requirements. Owing to the exceptional amount of work caused under the provisions of the Naval Defence Act, both the usual modes of procedure have been departed from in the case of the *Grafton*, and under Admiralty supervision she has been made almost ready for the pennant by her contractors. Save for her guns and torpedo tubes, she is practically in a condition for going to sea; she is probably more advanced than any recently-built contract ship has been at the time of her delivery.

So much attention has been taken up by the battleships of the *Hamilton* programme that the cruisers have received comparatively small attention. Notwithstanding, the cruisers are remarkable vessels, even in these days of advanced engineering. The Thames Iron Works secured contracts for three of the latter vessels, and have thus kept the historic Blackwall Yard in pretty full swing for some time past. The chief of these vessels was the *Blenheim*, a sister ship to the *Blake*. These two are of the largest class of cruisers yet constructed, being no less than 9,000 tons displacement and 375 feet in length. The *Grafton* is a smaller craft, although still a first-class cruiser. She was handed over to the government authorities by the contractors and taken to Chatham for completion about the middle of February.

Since her launch, which took place at the beginning of this year, the *Grafton* has been in the Victoria Dock, in company with her sister ship, the *Theseus*, since the latter vessel was launched a few months ago. The following are the chief details of the *Grafton's* design: The length is 360 feet; breadth, 60 feet; depth, 23 feet 6 inches; displacement, 7,350 tons; indicated horse power, 12,000. The machinery is by Humphrys, Tennant & Co., that of the *Theseus* being by Maudslay. The vessel is twin-screw, and the engines are of the triple-expansion type with cylinders 40 inches, 59 inches and 80 inches in diameter, by 4 feet 3-inch stroke. There are eight boilers 16 feet in

diameter and 9 feet 10 inches long, the working pressure being 155 pounds to the square inch. The boilers are single-ended.

The chief armament consists of two 9.2-inch guns, one placed forward and one aft on the upper deck and on the center line. These are protected by shields. There are to be ten 6-inch quick-firing guns, and 16 quick-firing guns of smaller caliber. There are also two above-water torpedo rooms, one forward and one aft. These are placed on the protective deck. There is also an under-water torpedo room. The torpedoes, we believe, are of the new 18-inch pattern. The protective deck has a mamimum of 5 inches on the slopes, and tapers to 2 inches fore and aft. The 5-inch thickness is made up of two thicknesses of 1¼-inch plates, and one thickness of 2½-inch plates.

A feature in which the *Grafton* differs from some other vessels is the forced-draft arrangement. Air locks are placed on the upper deck, so that the trunk leading to the stokehold is under pressure. There is a light grating over the top of the boilers, so that the air pressure extends above to the armored deck.

In the present condition of the ship the casemates in which the 6-inch quick-firing guns on the main deck are mounted are well shown. These casemates are armored chambers formed by a curved plate which projects rather less than 2 feet beyond the side of the ship. This curved front is of 6-inch steel armor. It forms a shield for the 6-inch gun, and the curve is such that the ends of the plate are about 3 feet 6 inches inboard. From these ends the sides of the casemate incline towards each other—the walls themselves being vertical—for a distance of 14 feet, and here they are 5 feet apart, that being the length of the back of the chamber.

To form the port through which the gun fires the plate has had an **I** slotted in it, and the two leaves thus made in the metal have been bent back until there is a distance of 3 feet 6 inches between their edges. The curve formed by bending back these edges has an arc of 2 feet measured on the inside of the plate, and the hollow thus formed affords, perhaps, the safest place to be found on the deck. The chord of the curve formed

by the front plate is 22 feet. There are four of these casemates on the main deck, two on each broadside.

The arrangements for supplying these casemates with ammunition have been happily thought out so as to get the greatest amount of protection in transit from the magazines to the guns. The ammunition is brought from the magazines, which are situated at the lowest point in the ship, on to the armored deck, and is then conveyed along an ammunition passage to the ready magazine which is situated below the casemates and is protected by 3-inch-steel armor; is, in fact, a 3-inch armored room 5 feet by 5 feet, but with the corners of the outer side rounded off, so as to present a curved surface to the impact of shot. There are three of these on each side of the ship on the protective deck. For serving the 6-inch guns on the upper deck an armored tube is provided, up which the ammunition is carried immediately to the gun by means of hand-worked lifts. The ammunition for the 9.2-inch guns is taken from below the armored deck up to the gun positions through armored tubes.

As is well known, the housing position for the broadside quick-firing guns is not inboard, the long muzzle projecting outwards at an angle which does not tend to the easy maneuvering of the ship in going alongside. Gear is therefore provided for dismounting the gun from its mounting, it being impossible to run in the whole apparatus, as in the old Scott carriage or the still older trucks; in fact, it is a necessity of the quick-firing principle that the mounting should be fixed to one spot. Gear is therefore provided for readily dismounting the gun and stowing it in an inboard position. This consists of hand gear and an overhead railway attached to the deck above. This gear has been neatly worked out, and is worthy of attention.

The fire service is a notable feature, the red color with which all the pipes and fittings are painted singling it out amongst the labyrinth of pipes, wires, and various devices with which a modern war vessel is crowded. Another feature worthy of note is the arrangement for warming the main deck, which is the mess deck. This consists simply of a number of ordinary stoves

of a description similar in appearance to the well-known "tortoise stove."

The conning tower is placed forward, and has 12-inch armor. The speed of the *Grafton* is to be 19.75 knots. She is designed to carry the large coal supply of 850 tons, which it is estimated will carry her 10,000 nautical miles at 10-knot speed. The cost is to be $1,680,000 for hull and machinery.

Blake.—In a recent discussion before the Institution of Naval Architects, Mr. W. H. White corrected a statement that the *Blake* could make only 19.12 knots. He said that she had made a run at this speed, but it was while she was expending 3,000 to 4,000 I.H.P. on her thrust bearing. The *Blenheim*, a sister ship, had made 20.5 knots all the way from the Thames to Portsmouth. The speed of the *Blake*, with natural draft, was nearer 21 knots than 19; with forced draft, her speed was 21.6 knots.

Blenheim.—To the editor of "Engineering." Sir: Permit me to make a brief statement supplementing the tabulated results for cruisers appearing on page 230 of your issue of February 24. [Page 504 of this issue of JOURNAL.]

The figures for speed appearing against the *Blenheim* do an injustice (which was no doubt unintentional) to that vessel. They correctly represent the results that have been published, but, standing without comment, they are misleading.

First, as to the performance under *natural draft.* The trial was made between Sheerness and Portsmouth in January, 1892, and extended over eight hours. During a considerable part of this time the ship was running in comparatively shallow water. I need not repeat details given in a paper read before the Institution of Naval Architects last year and afterwards published in your columns. Suffice it to say that the mean speed for the eight hours, taken by log, was 20.4 knots; but that this mean was very seriously affected by the "drag" when running in shallow water. For the two hours which the ship was running in deep water the speed was 21.3 knots by log, with 15,500 horse power and 96.5 revolutions.

I was careful to allude to the possible inaccuracies of log measurements in the paper just mentioned, but the trials of the *Edgar*

and subsequent trials of the *Blenheim* on the measured mile, have placed it beyond doubt that, with the natural-draft power, which can be maintained for eight hours, the *Blenheim* can steam 20½ to 21 knots in deep water. In the design, 20 knots was the speed named for natural draft. It will be remembered that the *Edgar* lost *three-quarters of a knot* in speed on the shallow measured mile at Stokes Bay, as compared with her running in deep water.

Second, as to the *forced-draft* trial of the *Blenheim*, made in October last.

This trial was made entirely for the purpose of testing the boilers and machinery under the maximum contract conditions after the boiler tubes were ferruled, and was most successful.

There was no intention to test the speed by accurate measurement. Your table gives 21.6 knots, which is simply a log speed, and obviously much below the truth.

The definite information obtained on this trial was that the average horse-power indicated for four hours was 21,400 horse-power, and the average revolutions per minute 105.3. Fortunately, we have been able to determine by runs on the measured mile at Stokes Bay the relation between revolutions of screw and speed of ship from 8 up to 18 knots. At the latter speed the "drag" of shallow water was becoming marked, and, after our experience with the *Edgar*, it was worse than useless to run the *Blenheim* at any higher speed on the mile. Besides these measured-mile results, we have the detailed observations made on the eight-hours' trial, and on the trials off Plymouth. In deep water 96.5 revolutions gave the ship a speed (by log) of fully 21 knots. For the four hours, the average of 105.3 revolutions, if the slip of screw remained the same, would, on this basis, give a speed of 23 knots, as against the log speed of 21.6 knots. There is no reason for anticipating any serious increase in slip in passing from 96 to 105 revolutions; and it will not be disputed that the revolutions of the screw furnish a far better approximation to the ship's speed than a small log towed in her wake.

Hereafter, when the *Blenheim* is on service, opportunities will occur for testing her speed under more exact conditions than have yet been possible, owing to the fact that we have no measured

mile with a depth of water proportioned to her high speed. But it will be admitted, I think, in view of the foregoing facts, that the vessel has more than fulfilled the estimated speed of 22 knots under forced draft.

So far as can be seen, the true deep-water speed corresponding to the forced-draft performance should be 22 knots per hour for four hours' continuous steaming.

I am, sir, your obedient servant,

W. H. WHITE.

March 2, 1893.

Cambrian.—On January 30th there was launched at Pembroke Dock-yard her Majesty's ship *Cambrian*, one of the twenty-nine cruisers for which provision was made in the Naval Defence Act. The displacement of the *Cambrian* when fully equipped and ready for sea will be 4,360 tons. She is 320 feet long, her extreme breadth is 49 feet 6 inches, and her depth of hold 15 feet 6 inches. When completed and ready for commission, with ammunition, stores, and all equipments on board, her mean draught will be 19 feet. Her upper deck will be 14 feet 3 inches and the center of her midship guns 18 feet above the water-line. The hull of the vessel, which is built of $\frac{3}{4}$-inch and $\frac{1}{2}$-inch steel plates, is sheathed with wood, $3\frac{1}{2}$ inches thick, to the height of about 1 foot above the water-line. A protective deck of varying thickness extends throughout the whole length of the ship. Her circular conning tower, situated on the forecastle, is formed of 3-inch steel armor plates, the top or covering plate being 1 inch in thickness; her torpedo director tower, situated nearer the stern, is wholly built of $\frac{1}{2}$-inch steel plates. Her engines have been manufactured by Messrs. Hawthorne, Leslie & Co., of Newcastle-on-Tyne. Like her sister vessels, she is to be fitted with twin, vertical, triple-expansion engines, the diameters of the cylinders being—high-pressure, 33 inches; intermediate pressure, 49 inches; low-pressure, 74 inches. The engines are to make 140 revolutions a minute, and to develop, under forced draft, during a continuous sea trial of four hours, 9,000 H.P., and with ordinary draft, 7,000 H.P.

Circe, *Alarm* and *Leda.*—These vessels were constructed under the Naval Defence Act from the designs of Mr. W. H. White,

C.B., Director of Naval Construction, at Sheerness Dockyard. Their principal dimensions are as follows: Length, 230 feet; breadth, 27 feet; displacement, 810 tons; draught, 8 feet 9 inches, and they are equipped with two 4.7-inch and four 3-pounder quick-firing guns, besides torpedoes and machine guns. The machinery was built by Messrs. John Penn & Sons, Greenwich. The engines are of the triple-expansion, vertical, twin-screw type, having cylinders 22 inches, 34 inches and 51 inches in diameter respectively, all with 21 inches stroke. The high-pressure cylinders are placed forward, and are fitted with piston slide valves, the intermediate-pressure and low-pressure slide valves being of the double flat-valve type and fitted with Church's relief arrangement. The cylinder covers and pistons are of cast steel, and the steam-chest doors of cast iron. The cylinders are carried on round forged-steel columns both back and front, the motion bars being carried from the cylinder bottoms and stayed to the columns, and the engine is well stiffened with diagonal stays. The bed plates are of cast steel, strongly secured to bearers built in the ship. The piston and connecting rods are of Siemens-Martin steel, the piston rods being fitted with combination metallic packing.

The crank shafts are hollow and made of steel by the Dennystown Forge Company, the cranks being set at 120 degrees apart. The surface condensers are made entirely of brass, the total cooling surface being 4,000 square feet. The circulating water is supplied by two 9-inch centrifugal pumps, made by the engine contractors. These pumps are arranged to pump out the bilge in the event of a leak at the rate of 300 tons per hour for each pump. The reversing gear is of the ordinary link-motion type, with solid bar links and adjustable working parts. Both steam and hand reversing gear are fitted. The air pumps are entirely of brass, and are worked from the intermediate pressure piston-rod crosshead. They deliver into a feed tank which overflows into the ship's reserve fresh-water tanks. A Kirkaldy's distiller, in combination with a Weir's evaporator, is furnished for each engine room, making together 75 gallons of fresh water per hour.

The propeller shafting is made of hollow steel 7¼ inches in diameter inboard and 8½ inches in diameter outside the ship. The boilers are four in number, of the marine locomotive type, with barrels 7 feet 6 inches in diameter and 15 feet 9 inches over all in length; they are of Siemens-Martin steel throughout, and are designed for a working pressure of 155 pounds per square inch, and a proof pressure of 245 pounds per square inch. The furnaces are eight in number, the heating surface being 6,250 square feet, and the grate area 182 square feet. The main steam pipes are of 6 inches diameter, and are wound with copper wire for greater security.

The official eight-hours' natural-draft trial of the *Circe* took place off Sheerness on the 27th of February, when Mr. Oram attended on behalf of the Admiralty, Mr. Moon representing the steam reserve, Mr. Pattison the dockyard, and Mr. J. P. Hall represented the engine contractors. The indicated horse-power developed was 2,620 with 223 revolutions of the engines, and .8 inch of air pressure in the stokeholds, the boiler pressure being 140 pounds, the vacuum 27 inches, and the speed by patent log 18.27 knots. The trial passed off without a hitch, and was considered highly satisfactory. On the 1st of March the official three-hours' full-power trial was successfully carried out in most unfavorable weather, there being a heavy sea running all the time. The indicated horse-power developed was 3,590 with 249 revolutions and 1.8 inches of air pressure in the stokeholds.

Repulse.—The contract trials of the engines of the new battleship *Repulse* were carried out during the last week of February, off Portsmouth, with very satisfactory results. The *Repulse*, which is one of the eight first-class battleships allowed for under the Naval Defence Act of 1889, was built at Pembroke dockyard and is engined by Messrs. Humphrys, Tennant and Co. The same firm, it will be remembered, supplied the engines for the *Royal Sovereign*, and has the contract for the engines of the *Hood* and the *Empress of India*.

The eight hours' natural-draft trial took place on the 28th of February; Mr. Soper representing the firm and Mr. Butler, R. N., the Admiralty. The vessel, which, when fully equipped, will

have a displacement of 14,150 tons, was not loaded to the full draught of water, as, instead of drawing 27½ feet, she drew 25 feet 9 inches. It was not, however, considered necessary by the Admiralty to insure the full draught, in view of the success of the *Royal Sovereign's* trials, and the contractors were chiefly interested in obtaining the necessary horse-power. The results were entirely satisfactory, and were as follows: Steam in boilers, 152.5 pounds; vacuum, 28½ inches in both condensers; revolutions, 99.4 and 99.3 per minute; indicated horse-power, 4,967 starboard, 4,566 port; collective horse-power, 9,533. The contract was exceeded by 533 horses, and throughout the trial the engines worked smoothly. The mean speed by patent log was 17.78 knots, but at full draught of water the maximum natural-draft speed is not expected to exceed 17 knots.

The forced-draft trial was carried out on the second of March, and proved equally successful. The results were as follows: Steam in boilers, 150.4 pounds; vacuum, 28½ inches; revolutions per minute, 104.1 starboard, 103.8 port.

Indicated Horse Power.	Starboard.	Port.
High	1,597	1,603
Intermediate	2,175	2,009
Low	2,016	1,902
Total	5,788	5,514
Collective indicated horse power	11,302	

The mean air pressure employed under natural draft was .45 inch, and under forced draft .91 inch. At the second trial the speed by log was increased, and averaged during the four hours' run 18.2 knots. The consumption of coal under natural draft was 2 pounds per indicated horse power. On the conclusion of the forced-draft test the usual trials of the steam-steering gear were carried out. Although these trials, insisted upon by the Admiralty, are of no practical value, it may be interesting to note the following results: From amidships to hard-a-port, 10 seconds; hard-a-port to hard-a-starboard, 37 seconds; hard-a-starboard to hard-a-port, 30 seconds, and from hard-a-port to amidships, 14 seconds. The contract power required was 9,000 under natural and 11,000 under forced draft. The *Repulse* has now been taken back to the basin to have her machinery opened up

for final examination, and during the next few days the Admiralty will no doubt accept the engines from the firm.

Vulcan.—The torpedo store-ship *Vulcan*, which has caused so much anxiety to the Admiralty since her completion last summer, and which was regarded only a few months ago as an absolute failure, was subjected to a forced-draft trial in March, which passed off in a highly satisfactory manner. The vessel was laid down at Portsmouth Dockyard in the summer of 1888, and was completed early in 1892. She was designed to be an improved type of torpedo dépôt ship, and to take the place of the *Hecla*, which was originally built for the White Star Line and bought for the Naval service. Beyond all doubt the *Vulcan* is a well-designed ship, but although the construction of her engines and boilers was intrusted to the well-known firm of Humphrys, Tennant & Co., the Admiralty designers insisted that she should be supplied with the Whitehall double-ended boilers, and this has been the chief cause of her failure to develop the contract horse-power. The Admiralty acknowledged that a blunder had been committed, and for the past six months naval engineers have been endeavoring to overcome the inherent weakness of her double-ended boilers. Whatever may be thought of the Admiralty ferrule, it must be acknowledged that this device has served to pull many a ship through her trials, and their lordships ought to be very grateful to the inventor. In the case of the *Vulcan*, every expedient was tried with equal lack of success, until, as a last resort, the tubes were ferruled, with the pleasing result that during the trial the boilers withstood 1.8 inches of air pressure without a sign of leakage in the tubes. The trial lasted four hours, with the following results:

	Pounds.	
Mean steam in boilers	148	
Vacuum (starboard)	26.8	
Vacuum (port)	26.2	
Revolutions	99.2 and	99.7
Indicated Horse Power.	Starboard.	Port.
High	1,690	1,871
Intermediate	1,949	1,922
Low	2,295	2,305
Total	5,934	6,098
Collective indicated horse-power	12,032	

The original contract was for 12,000 horses under forced draft. The vessel averaged a speed of 20.2 knots by patent log, her extreme draught of water being 24 feet. [For very complete descriptions of the *Vulcan* see vol. IV, pp. 820–829.]

Renown.—This new battleship, which is being constructed at Pembroke Dockyard, from designs by Mr. W. H. White, Director of Naval Construction, was mentioned but not described by Lord George Hamilton last year. Lord Spencer has given some further particulars, but nothing like a detailed description of the methods by which it is proposed to give her armored or other protection. Her normal displacement will be 12,350 tons, or 1,850 tons more than that of the *Centurion*, and 1,800 tons less than that of the *Royal Sovereign*. On this displacement she will carry four 10-inch B.L. guns in two barbettes, mounted similarly to those in the *Centurion*, and a secondary armament of ten 6-inch quick-firers, and 20 small quick-firing guns. The barbette armor will be thicker than that of the *Centurion*, and the system of protecting the secondary battery will be similar to that adopted with the 6-inch quick-firers of the *Royal Sovereign*.

The *Renown* will be wood sheathed, provided by contract with engines of 10,000 horse power, to give a speed of 17 knots on the eight-hours' natural-draft trial, and a maximum speed of 18 knots with moderate forced draft.

Ramilies.—This first-class battleship of the *Royal Sovereign* type, which was recently received from the Clyde, where she was built and engined by Messrs. Thomson, made an eight hours' contractor's trial of her engines, at Portsmouth, on April 25th, under natural draft. The ship was designed for a mean load draught of 27 feet 6 inches, and her mean immersion on trial was 25 feet 2 inches. The average steam in boilers was 149 pounds, which was maintained with the remarkably low air pressure of .22 inch. The vacuum was very good, averaging 27.8 inches, and the port and starboard engines were worked regularly at 96.7 revolutions. Under these conditions the engines developed 4,705 and 4,710 I.H.P. respectively, thereby securing a collective I.H.P. of 9,415 and a log speed of 16.75 knots. The designer's estimate was 9,000 H.P. and a 16-knot speed. The coal con-

sumption during the run was 1.7 pounds per I.H.P. per hour, which was much below the average.

Majestic and *Magnificent.*—These two battleships of the new programme will be built at Portsmouth and Chatham respectively. They will exceed the *Royal Sovereign* in tonnage, and will be 390 feet in length with a beam of 75 feet. They are to be armed with four 12-inch, 46-ton guns, twelve 6-inch and twenty-eight smaller quick-firing guns. Their machinery of 13,000 H.P. to give 18 knots, forced draft, and 16½ knots, natural draft, will be built by contract. While in general appearance these vessels will somewhat resemble the *Royal Sovercign* class, it is anticipated that they will show considerable modification in the principle of giving armored protection.

Terrible and *Powerful.*—Of these two first-class cruisers which are to be put out to contract, nothing very definite is yet known. Questioned upon this matter, the Secretary of the Admiralty, in the House of Commons, and Lord Spencer, in the Upper House, have refused to give particulars, alleging that it is more politic not to give the information until the arrangements are so fully complete as to obviate the chance of foreign Governments forestalling us. This attitude is generally looked upon as an admission that the designs of the vessels are not yet ready, since it is well-known that if they were, foreigners would be pretty sure to obtain them, despite all precautions. All that Lord Spencer could be made to say, was, that they are to embody the results of experience gained with our existing cruisers, and in speed, coal supply, armament and protection, to surpass anything of the kind built and building. The heaviest cruiser building is the Russian *Ruric*, of 10,923 tons displacement; the fastest, the Argentine *Nueve de Julio*, which made a mean of 22.74 knots with forced draft; the United States *Columbia* can stow 2,000 tons of coals; while the French *Dupuy de Lôme* is completely covered with armor. There is, therefore, ample scope for the talent and ingenuity of the Admiralty designers. That the new vessels will be battleships in everything but name, is also a matter of very little doubt. [English Exchange.]

The new second-class cruisers.—Three of these vessels are to be

built in the Government dockyards. They are improvements upon the *Astræa* class described below, and will be of 5,500 tons displacement, 350 feet length, 53 feet beam, and 20 feet mean load draught. The engines, which are also to be built in the dockyards, will be of 9,600 horse power at forced draft, giving a maximum speed of 19.85 knots, and 8,000 horse power at natural draft, with a speed of 18.5 knots. They will each carry five 6 inch, six 4.7-inch, and nine smaller quick-firing guns. The *Talbot* will be built at Devonport, the *Minerva* at Chatham, and the *Eclipse* at Portsmouth. What is particularly noteworthy about these new cruisers is their length, some 30 feet more than that of the *Astræa* class. It is probable that a greater sea speed is anticipated as a result of this increased length, for it is well known that short ships cannot be made, except at great expenditure of power, to maintain a pace in heavy or even moderately rough water.

Barham.—This vessel, which has been the cause of much anxiety, has at length made satisfactory steam trials. The eight-hours' trial was made April 12th, the tubes of her boilers having been previously ferruled and the ship fitted with a new set of fans in place of those which failed at the last trial. The engines developed 3,589 I.H.P. and a log speed of 18.91 knots, the average pressure in the boilers being 131 pounds, and the air pressure 1.3 inches. On April 15th a forced-draft trial took place, when, for three hours, the vessel averaged 20.8 knots by patent log, developing a horse power of 4,538, which is the highest speed and power she has yet obtained. The pressure in the boilers was 137.5 pounds, with an air pressure of a little over two inches. This is a very successful demonstration of the value of the ferruling process, the credit for which is given to Mr. Edward C. Peck, of Messrs. Yarrow's firm at Poplar, and to Mr. Oram, R. N., of the Admiralty.

The Torpedo Flotilla.—Ten boats of 140 feet in length, now being constructed, should compare favorably, as regards speed, habitability and handiness with anything of their kind; they will all be ready for trial in a short time. Of the 20 torpedo-boat destroyers of 27 knots speed, six are already in hand, and

will be completed during the financial year. Messrs. Thornycroft will have the *Daring* ready for trial in July next, and the *Decoy* in August. Messrs. Yarrow will follow with the *Hornet* and *Havoc* in October, and Messrs. Laird will turn out the *Ferret* in February, 1894, and the *Lynx* a little later. The length of these new vessels will be 180 to 185 feet, beam 18.5 to 19 feet, and draught about 5 to 6 feet. The proposed armament is one 12-pounder, new Elswick pattern, three 6-pounder quick-firing guns, and three torpedo-ejecting tubes. The engines will be about 3,500 H.P., and at normal draft the coal capacity about 60 tons. At present it has been decided to give them crews of 40 officers and men. Although 27 knots is spoken of as the maximum rate, a sea speed of from 23 to 24 knots is all that is anticipated and hoped for in the first vessels finished, but as torpedo boats of 30 knots speed are already projected, it is obvious that before the twenty destroyers are completed the standard of speed will have to be raised considerably.

The New Sloops.—These are to be named the *Torch* and *Alert*, and will be built at Sheerness, engines included. They are to be improvements upon the *Goldfinch* class of gun vessel, and will be used on foreign stations. They will be 960 tons displacement, 180 feet in length, 32 feet 6 inches beam, and 11 feet 6 inches mean load draught. Engines of 1,400 H.P., with forced draft will drive them 13.25 knots, and 1,050 H.P. natural draft will give a sea-speed of 12.25 knots. The armament will comprise six 25-pounder and four 3-pounder quick-firing guns.

Astræa.—The *Astræa*, recently launched at Devonport, was designed by Mr. W. H. White. She was laid down August 14, 1890. Her dimensions are: Length, 320 feet; breadth, 49½ feet; mean load draught, 19 feet. The weight of her hull, armor and backing will be 2,460 tons, with a total displacement of 4,360 tons at load draught. Her coal capacity is 400 tons. Her armament, which will cost $54,000, will consist of two 6-inch and eight 4.7-inch quick-firing guns, eight 6-pounders and one 3-pounder Hotchkiss, and four 5-barrel .45-inch Nordenfelt machine guns; in addition to which four torpedo tubes are to be fitted, viz: one stem, one stern and two broadside tubes. The

money to be spent on the construction of the *Astræa* this year is $345,000, and when completed the total estimated cost of the ship, including guns, will be $1,250,000. This is a revised estimate, the original having been $1,210,000. Although the *Astræa* is a second-class cruiser, she is to be fitted as an admiral's ship. The machinery, which has been manufactured at Keyham, is ready to be placed on board. The propelling machinery will consist of two sets of triple-expansion, surface-condensing vertical engines, capable of developing a collective force of 9,000 horse power when under forced draft and 7,000 horse power with natural draft. The principal dimensions of the main engines are: Cylinders, high pressure, 33 inches; intermediate pressure, 49 inches; low pressure, 74 inches in diameter, with a stroke of 3 feet 3 inches. The propellers are of the three-bladed type, and will make 140 revolutions per minute, which will give an estimated speed of 20 knots per hour. The boilers are eight in number, of the single-ended return-tube type; they are constructed entirely of steel, and will work at a pressure of 150 pounds per square inch, each boiler having three furnaces of Fox's corrugated type. The auxiliary machinery includes feed, fan, fire and bilge, main and auxiliary circulating, turning, starting and workshop engines, and steam injectors for filling reserve fresh-water tanks.

Sybille.—The cruiser *Sybille* went outside Plymouth Breakwater a few weeks ago for a few hours' natural-draft trial of her machinery. This trial was for the purpose of proving the efficiency of a patch put on one of the wing furnaces in the place of a defective portion which had been removed. The results of the trial were as follows: Mean steam in boilers, 149 pounds; vacuum, starboard, 29 inches; port, 28.3 inches; revolutions, starboard, 132.5; port, 131.9; air pressure, .19 inches; indicated horse power, starboard, 3,655; port, 3,595; total, 7,250; speed by log, 18.98 knots per hour. [For description, see p. 304, vol. IV.]

FRANCE.

Descartes.—This vessel is a second-class sheathed cruiser, intended for distant service. The hull dimensions are 316 feet by

42 feet by 18.8 feet; displacement, 3,950 tons; speed under forced draft, 19 knots. There is a double bottom extending throughout nearly the entire length. The protective deck is 1.6 inches thick on the slopes and 1 inch on the flat. The sheathing is in two sections, $2\frac{3}{4}$ and $2\frac{3}{8}$ inches thick respectively, which reach to a height of about three feet above the water line, and it is secured to the plating by galvanized steel bolts.

There are two military masts with ladders on the inside, each having three tops. The lower is for the officer in command, the middle one for 47 mm. R.F.G., while the upper is for 37 mm. R.F.G., and search light. A conning tower is fitted forward with plates 2.75 inches thick on the forward side, and 1.6 inches thick on the after side.

The engines are twin-screw, vertical and triple-expansion, and there are to be sixteen Belleville boilers arranged in four fire rooms. The coal supply at the designed displacement is 550 tons.

The contract trials are very elaborate and include—

(1) A four-hour trial under forced draft, during which the I.H.P. must not be less, as an average, than 8,500 and the speed 19 knots.

(2) A six-hour trial under natural draft for 18 knots.

(3) A twenty-four-hour trial to determine the ability to maintain a good performance, when the I.H.P. must be about 5,500.

(4) Two trials for economy, one at full power and the other when making 10 knots.

The armament comprises four 16-cm. (6.3 inches) B.L.R., ten 10-cm. (3.94 inches) B.L.R., fourteen 47-mm. R.F.G., eight 37-mm. R.F.G., and two torpedo tubes.

Fleurus.—On March 18th the torpedo cruiser *Fleurus*, for the French Navy, was launched at Cherbourg. The vessel, which is a kind of modified *Condor*, is 216 feet long, 29 feet 2 inches beam, and 15 feet 6 inches deep, with a displacement of 1,300 tons. The vessel will be fitted with engines of 4,000 horse power, by which a speed of over 19 knots is expected to be attained. The armament of the vessel will comprise five 10-cm. quick-firing guns, two 65-mm. quick-firing guns, and four torpedo tubes.

SWEDEN.

Thule.—The new Swedish ironclad, the *Thule*, which is the largest Swedish warship of modern type, has lately been launched at Finnboda, Sweden. She has a good many points in common with the *Svea* and *Göta*, built at the Motala establishment, but is larger and has, what the others have not, a ram. The principal dimensions are: Greatest length, 271 feet; length on water line, 267 feet; greatest breadth, 48.7 feet; and the draught, full armament, is 16 feet; displacement, 3,165 tons. The space between the outer armor and the inner side of the vessel is divided into cells, 50 on each side, and 86 in the bottom, and the ship is, of course, divided into watertight compartments. The citadel is protected by armor varying in thickness between 12 and 8 inches. There are two turrets. The one for the heavy guns has a diameter of 23 feet; height, 6 feet, and can revolve 292 degrees. It contains two Armstrong guns of 25.4 centimeters caliber, placed parallel with each other. The heavy armor plates for the turrets and sides have been delivered by Creusot. The engines, powder magazines, &c., are protected. Otherwise the armor plates are of Swedish make from Domnarfolt. Inside the armor is a substantial coating of teak. The two propellers are worked by engines of 3,200 I.H.P. The normal speed is 15 knots, but it can be raised to 16 knots. The ram, of Bafors steel, weighs 10 tons without the heavy double armor. There are two movable torpedo tubes. In addition to the two guns mentioned, there are four Bafors 15-centimeter guns, and five rapid firing Maxim-Nordenfelt 57-millimeter guns, which are placed on the upper deck, together with some Nordenfelt machine guns.

MERCHANT STEAMERS.

THE NEW SHIPS OF THE INTERNATIONAL NAVIGATION COMPANY.

These consist of four twin-screw steel steamships to be built by the William Cramp & Sons' Ship and Engine Building Company. Since the passage of the act of March 3, 1891, providing for an ocean mail service, a great deal has been said about the American ships to be built by the owners of the *Paris* and *New York*. Now that the material for hulls and boilers has been contracted for, and work on these boats has been started, it is expected, however, that information in greater detail can be obtained shortly. The general features of both hull and machinery are settled.

It is not the intention of the International company to build five boats at present, as has been erroneously stated a great many times. Only four boats for the Red Star line are to be built, and they are to be but 510 feet between perpendiculars, or 17 feet shorter than the *Paris* or *New York*. The first of these four steamers will be put on the Southampton route, temporarily, with the *Paris* and *New York*, while the other three will run to Antwerp in the Red Star line. A fifth boat, the "world beater," as she is now styled, will be constructed later, but it can be said with certainty that plans for this steamer have not as yet been considered, as it is the intention of the International company, as well as the Cramps, to take advantage of all improvements shown in the new Cunard liners *Campania* and *Lucania* and the White Star steamer building at Belfast, Ireland. This fifth boat is, of course, intended for the Southampton route with the *Paris* and *New York*, and when completed will replace the first of the four boats now under contract.

Dimensions of each of the first four boats, as now fully agreed upon, are as follows: Length between perpendiculars,

510 feet; breadth of beam, moulded, 63 feet; depth, moulded, 42 feet. The hull is to have a double-bottom constructed on the cellular principle, sub-divided by athwartship bulkheads and a longitudinal division arranged for heeling purposes, the whole to be available for water ballast. The stem will be straight and the stern elliptic, and there will be three pole masts.

There will be promenade, saloon, upper, main, and orlop decks, the three first named to be plated from end to end. The main deck will be plated for the length of the machinery space, and will have stringers and ties beyond. Wood decks will be laid on all decks. The promenade deck will extend unbroken the whole length of the ship. On 26 feet draught of water, the vessel is to carry 1,900 gross tons dead weight cargo, and 3,000 tons of coal, baggage, stores, etc.

The engines will be the most powerful quadruple-expansion marine engines in the world. They will be required to develop about 7,500 horse-power each. The cylinders will be 36, 50, 71, and 100 inches by 60 inches stroke, two sets of engines being placed in each boat to turn the twin screws, which will be built-up with three blades. Steam will be furnished at 210 pounds pressure by six double-ended steel boilers, 20 feet long and 15 feet 7½ inches diameter, the battery to have 48 Purves furnaces, 39 inches in diameter, and to be fitted with Serve's patent tubes. The total grate surface will aggregate 820 square feet, and the heating serface about 30,000 square feet.

Piston valves will be fitted throughout and operated in the usual manner. The crank-shafts, eccentric straps, and connecting rods and piston-rods will be of forged steel. The valve gear will be of the link type, controlled by a steam cylinder, and also by an auxiliary hand gear. Many features will be added to insure the perfect working of these engines.

The columns will be of cast steel, forked at bottom. The thrust blocks will be of the usual horseshoe type, and the thrust shafts are to be about 13 feet long. The line shafting will be of forged steel, the bearings to be of cast iron. The air pumps will be driven from the main engines, but the circulating pumps will be independent.—*Marine Review.*

STEAMSHIP *MANITOU*—LAKE MICHIGAN AND LAKE SUPERIOR TRANSPORTATION COMPANY..

Another of the transportation companies, the Lake Michigan and Lake Superior Company, of Chicago, which has been operating passenger and freight steamers on Lakes Michigan and Superior for a great number of years past, will add still another ship to its fleet upon the opening of the passenger season. The steamer will be a most costly and elegant modern passenger ship and will be named *Manitou*. The Lake Michigan and Lake Superior Transportation Company is probably the oldest concern on the Lakes maintaining an independent line of passenger and freight steamers, and under the undivided and strict attention that has been given to its passenger business, it has grown to such large proportions that, in order to accommodate its increasing patronage during the tourist season (which usually comprises the months of July and August and parts of the months of June and September), another boat specially fitted to this service has been planned with a great deal of care.

The new steamship now building by the Chicago Shipbuilding Company is the result of plans that were under discussion many months, and will be one of the finest vessels ever seen on the Lakes, presenting many novel and agreeable features for the comfort, convenience and safety of the traveling public. This ship is being built of steel throughout, to the rules of the Bureau Veritas.

The ship has an over-all length of 295 feet, keel 275 feet, beam 42 feet, and depth to spar deck of 24½ feet, or to hurricane deck 32½ feet. The water botton is 3½ feet deep, extending the whole length of the vessel and divided into eight compartments, while seven water-tight bulkheads in the hull afford a degree of subdivision and safety which has never as yet been introduced in vessels of her class.

The machinery, being built by the Cleveland Ship Building Company, consists of a triple expansion engine with cylinders 23, 38 and 62 inches diameter and 36 inches stroke, and two steel gun-boat boilers, 11 feet diameter by 21 feet in length, placed in a separate compartment, connected to one oval smoke-

stack and carrying 160 pounds of steam. The ship is expected to make from 15 to 16 miles an hour easily under natural draft, and for forcing the speed at any time a novel arrangement of induced draft is being fitted, consisting of two Sturtevant exhaust fans placed in the base of the stack, through which, when in operation, the products of combustion will be drawn, thus increasing the draft and combustion of fuel.

Connected air, bilge, and circulating pumps, independent feed pumps and a feed-water heater are other features in the machinery. There will be three 44 inch furnaces to each of the boilers, and one donkey boiler, 4 feet diameter and 7 feet high. The pistons of the engines will be of cast steel, the same as is now generally used in the Navy. [Exchange.]

The *Isle of Arran* is a new paddle-wheel steamer built for service on the Clyde. The following are the principal dimensions: Length over all, 210 feet; breadth amidships, 24 feet; over paddle-boxes, about 47 feet. The machinery, while not novel in design, is of a type which has done good service in Clyde boats. Its first cost is comparatively small, and it is fairly economical. The vessel has a single diagonal engine with a cylinder 52 inches in diameter by 60-inch stroke. It is jacketed with high-pressure steam. The jacket is drained by being connected with a Weir's feed heater, where the feed water is heated to about 200 degrees Fahr. before being delivered to the boiler by one of Weir's automatie feed pumps. The water is pumped from the hot well to the heater by a feed pump on the main engine, worked from the air-pump crosshead. The main slide valve is double-ported, and cuts off the steam at $\frac{9}{16}$ of stroke. It is fitted with Thom's patent ports for increasing the compression by admitting a part of the steam, which is just about to exhaust from one end of the cylinder into the other end of the cylinder, where it is trapped and compressed by the returning piston. The slide valve also has the pressure on the back relieved by Thom's patent balanced cap, which experience has shown to work admirably. Lockwood and Carlisle's rings are fitted to the main piston. The piston-rod is steel and the guide frames wrought iron, with cast-iron shoes and white metal slippers. Under the guides, arranged

athwartship, is the surface condenser, made of steel plates and having the large cooling surface of 2,550 feet. The circulating water, which passes twice through the condenser, is supplied by a centrifugal pump and engine supplied by Messrs. Drysdale, of Glasgow. The discharge pipe is 12 inches in diameter, and a vacuum of 27 inches is maintained in the condenser. The reversing engine has a cylinder 13 inches in diameter by 21 inches stroke, connected to the main slide-valve spindle. There is a single loose eccentric on the main shaft, and the piston-rod of the reversing engine and the main slide-valve spindle are put in and out of gear in the usual way from the starting platform. The main shafts, which are of iron, are 12¾ inches in diameter, the bearings being 19 inches long, and the crankpin, which is of steel, 10 inches in diameter by 15 inches long. The paddle-wheels are 17 feet 3 inches in diameter over the floats, which are curved steel plates 9 feet 6 inches by 2 feet 10 inches by ½ inch thick. Steam is supplied from a single haystack boiler 14 feet 9 inches in diameter by 15 feet 3 inches high, loaded to 60 pounds pressure, having a heating surface of 2,945 square feet, and 130 square feet of fire-grate. The boiler is covered with a non-conducting substance recently introduced by Messrs. Hafter. It consists of briquettes of compressed granulated cork, fitted to the boiler and secured by wire, the whole being coated with composition. The indicated horse-power is about 1,300, with 46 revolutions. The cruising speed is 16½ knots.

BOOK REVIEWS.

"ENGINEERING" FOR APRIL 21, 1893.

It is hardly necessary to commend "Engineering" to any engineer who tries to keep posted in his profession, for it easily ranks first among the technical journals in the English language and probably in any language. In the number under consideration, however, the editors and publisher have outdone themselves and have produced a magazine which is the finest specimen of a technical publication ever issued.

Its special value to marine engineers and naval architects consists in the first section of the number—forty pages devoted to the description of the new Cunard steamers, *Campania* and *Lucania*. This is almost a treatise on practical ship and engine building and is profusely illustrated with valuable diagrams and drawings, besides nine two-page engravings of features of the vessels. Had this section of the number been published in book form it would have been considered cheap at many times its actual price.

The very full account, which appears elsewhere in this number of the JOURNAL, of the new Cunarders is only a brief extract from the account in "Engineering."

In addition, however, there is an elaborate illustrated account of the World's Columbian Exposition at Chicago, which is of the greatest value, as it was prepared by Mr. Dredge himself, who, as one of the British Commissioners to the Exposition, has given special attention to it.

We would advise every reader of the JOURNAL to procure a copy of this number of "Engineering," as it will prove a most valuable addition to his library. It can be procured in this country of John Wiley & Sons, 53 East 10th St., New York

City, who issue an edition on heavy calendered paper and bound in boards for $2.50, postage paid.

ELECTRICAL TABLES AND MEMORANDA.—BY SILVANUS P. THOMPSON, D. Sc., and EUSTACE THOMAS.

This is a vest-pocket edition of a collection of tables and data which will prove very useful to all who have to do with the practical applications of electricity. It does not deal with theory, but gives in very compact form just what one wants to know about electric lighting, power transmission, storage batteries, &c., &c. Naturally some points are not worked out in as great detail as would sometimes be desired, but, for practical everyday work, it covers its field admirably.

For sale by Spon & Chamberlain, 12 Cortlandt street, New York. Price, 50 cents.

BUSLEY'S NAVAL AND MERCHANT STEAMSHIPS.

The interest which Germany has taken in the development of her merchant marine and navy is well shown in a small volume lately published in Kiel, "Die Neueren Schnelldampfer der Handels und Kriegsmarine." The author, Professor Carl Busley, of the Royal Naval Academy at Kiel, has given a very interesting history of the growth of modern fast ships. The first chapter is devoted to a review of the development of ocean navigation; the second to a comparison of the requirements for the merchant service and the navy. Nine chapters follow, containing numerous excellent engravings and much valuable information, especially about the transatlantic mail steamers.

It is interesting to note many references to the ships built in the United States and to early river navigation. Even the first fast packets that plied between New Orleans and St. Louis, early in the forties, are mentioned. An entire chapter is devoted to triple-screw ships, and numerous cuts of the U.S.S. *Columbia* are given.

The tables, showing the growth of the German merchant service and the gradual decadence of our own merchant marine

during the past fifteen years, are not calculated to fill an American sailor with enthusiasm for the future.

The volume is handsomely printed and well worth perusal.

ALTERNATING CURRENTS OF ELECTRICITY.—THEIR GENERATION, MEASUREMENT, DISTRIBUTION AND APPLICATION.—By GISBERT KNAPP, C. E., WITH AN INTRODUCTION BY WM. STANLEY, JR. 1893. The W. J. Johnston Co., Limited, 41 Park Row, New York City. Price, $1.00.

Although the great field for alternating currents seems, at present, to be for long distance transmission, it is a development of electricity of great interest to all who desire to keep abreast of the subject. Unfortunately, most of the works treating of alternating currents have thus far been pretty hard reading for the average student.

In the present work the principles are developed in a simple manner that can be followed by any reader, and the various applications are sketched in a broad and instructive way that clears away the many mysteries with which they were supposed to be surrounded. The few mathematical formulæ in the book are confined to appendices.

The treatment throughout is thoroughly practical, and the data and discussion on the design and construction of apparatus will be found of great value to the electrician and designer. To the student and the general public this work will be a particular boon, bringing within their grasp a subject of the greatest importance and interest.

ALTERNATING CURRENTS: AN ANALYTICAL AND GRAPHICAL TREATMENT FOR STUDENTS AND ENGINEERS.—BY DR. FREDERICK BEDELL AND DR. ALBERT C. CREHORE. New York. The W. J. Johnston Company, Limited. 325 pages. 112 illustrations. Price, $2.50.

This is a more elaborate work than the little treatise by Kapp, already noticed. The subject is developed mathematically and analytically, but so gradually and logically that it is readily grasped by any one with a fair mathematical training.

The first fourteen chapters contain the analytical development, commencing with circuits containing resistance and self-induction only, resistance and capacity only, and proceeding to more complex circuits containing resistance, self-induction and capacity, and resistance and distributed capacity. A feature is the numerical calculations given as illustrations.

The remaining chapters are devoted to the graphical consideration of the same subjects, enabling a reader with little mathematical knowledge to follow the authors, and with extensions to cases that are better treated by the graphical than by the analytical method.

Parts of the book have previously appeared in various electrical journals, and have received flattering commendation.

CORRECTIONS AND ADDITIONS TO THE LIST OF MEMBERS AND ASSOCIATES OF THE SOCIETY.

MEMBERS.

Bartlett, F. W., P. A. Engineer, U. S. Navy.....Detroit, Mich.

Cowie, George, Jr., P. A. Engineer, U. S. Navy.....U. S. S. *Yantic*.

ASSOCIATES.

Guy, A. E.,.....Bureau Steam Engineering, Navy Department.

Livingstone, W. A., Marine Engineer, Secretary Michigan Navigation Co.....
.....12 Woodward Ave., Detroit, Mich.

McDermott, Geo. R., Naval Architect, Asst. Prof. Naval Architecture, Cornell University.....
.....Ithaca, N. Y.

Rowland, Thos. F., Jr., Secy. and Treas. Continental Iron Works.....
.....Station G, Brooklyn, N. Y.

SUBSCRIBER.

Heese, Lieutenant Albrecht, Military Attaché German Legation.....Washington, D. C.

INTERNATIONAL ENGINEERING CONGRESS.

Probably all of the readers of the JOURNAL have heard in a general way of this great assemblage of engineers which is to be held at Chicago in connection with the World's Columbian Exposition, July 31–August 5, of this year. It covers all the fields of engineering except electrical, which is to have a special congress.

The various branches of engineering are managed in divisions, most of which are cared for by the great National societies, but Military and Naval engineering have been entrusted to officers of the United States Government.

The division of Marine and Naval Engineering and Naval Architecture is managed by Engineer-in-Chief George W. Melville, Chief of the Bureau of Steam Engineering of the Navy Department, with the assistance of an Advisory Council and a Corps of Honorary Presidents. The latter consists of the gentlemen who now hold or have held positions corresponding to our own Engineer-in-Chief and Chief Constructor, in our own or in foreign navies. As will be seen by the appended list, the eminent men who have accepted the honor are known throughout the world for their professional distinction.

The Advisory Council is composed of eminent naval architects and marine engineers at home and abroad, and, as will be seen at once, it includes nearly all of the most prominent men in the profession.

With the advice and active co-operation of these two bodies, a series of papers by authors of marked ability and experience has been secured, and it should be added that the kind assistance of such societies as the Institution of Naval Architects, the Institute of Engineers and Shipbuilders in Scotland, and the Northeast Coast Institution, has also done much for the Division. Messrs. Maw and Dredge, the editors of "Engineering," and Mr. Dunell, of their staff, have also been of great assistance.

The list of authors and papers given below will at once satisfy all professional men that a rare treat has been provided, and, when it is understood that these papers are to be circulated confidentially in advance, so as to insure a careful discussion, it can be realized that the proceedings will form probably the most valuable collection of papers that has ever appeared on subjects connected with marine engineering and naval architecture. Those who have seen the papers thoroughly endorse what has been said, and, in fact, there are single papers whose value alone is more than the proceedings will cost.

The well-known publishing house of John Wiley & Sons, of New York, has, with patriotic spirit, undertaken the publication of the Proceedings and proposes to issue them in excellent style.

It is, of course, impossible to give the exact size of the work, but a conservative estimate places it at some 1,500 large octavo pages, including some 200 plates. It will be bound in one or two volumes, as is found most convenient when completed.

The price will be $10.00 for the complete bound volumes, which it is expected will be ready for delivery in October. Subscriptions may be sent at any time to Messrs. Wiley, at 53 East Tenth street, New York city, or to the Secretary of the Division, Passed Assistant Engineer W. M. McFarland, U. S. Navy, Navy Department, Washington, D. C.

It is urged upon all naval architects and marine engineers who expect to visit the Exposition at Chicago, to time their trip so as to attend the sessions of the Congress. Apart from the direct benefit professionally will be the pleasure of meeting eminent members of the profession, whose faces will hereafter be as familiar as their names are now.

The reception rooms and office of the General and Executive Committees of the Associated Engineering Socities of the United States and Canada, and of the General Committee of the World's Congress Auxiliary on the International Engineering Congress, on and after May 1st, 1893, and until the close of the World's Columbian Exposition, will be at No. 10 Van Buren street, Chicago, Illinois.

All communications for this Committee after May 1st should

be addressed to, or to the care of, Mr. Max E. Schmidt, Secretary. Visiting engineers may have their mail thus addressed.

The Committee Rooms of the Associated Socities at the Exposition will be situated in the northwest corner of the gallery of the Mines and Mining Building, where the Secretary or some of the staff will be present during the Exposition, to meet visiting engineers.

TIME AND PLACE.—The time assigned to the Engineering Congress is the week beginning Monday, July 31st, and ending Saturday, August 5, 1893.

There will be an opening general session at 10 o'clock on Monday, July 31, 1893, in one of the large halls of the Art Palace in the Lake Front Park in Chicago. This is in the business or down-town part of the city and not at the Exposition grounds. After appropriate addresses the divisions will be convened in their respective session rooms in the same building.

CARDS OF ADMISSION.—Cards of admission to the Engineering Congress will be issued by the Secretary of the General Committee, upon the presentation of introductory cards given by the officers in charge of the several divisions, or by the officers of the societies invited to participate in the Congress.

The cards of admission will entitle the holder to attend the meetings of any of the sessions. There will be no entrance fee for participation in the proceedings.

PROCEEDINGS.—The proceedings will consist of the reading and discussion of such papers as shall have been accepted by the management of each division, and according to the rules which may be adopted.

Correspondence relative to the time when proposed papers must be transmitted for acceptance should be addressed to the representatives of the divisions concerned.

Opportunities will be given for the display of wall diagrams and of stereopticon (magic lantern) views in illustration of papers.

It is intended that all papers printed by the divisions shall be in English. Papers may be offered in French, Spanish, German and other languages, and, if accepted, will be translated and printed in English. Discussions may be in any one of the three

languages mentioned, and interpreters will be provided whenever necessary. It is evident that contributions requiring translation should be transmitted at the earliest possible day. Information concerning scale, size of drawings, etc., should be obtained from the representatives of the divisions, who will also issue advance copies of the papers.

A daily programme will be issued each morning by the General Committee, stating the order of papers and proceedings, so that engineers in attendance may select the sessions at which they prefer to be present. This programme will indicate the rules to be followed, and, as far as practicable, the speakers who are to take part in the discussions.

There will be five morning sessions of the divisions, some of which may be joint sessions.

The Congress will terminate with a general morning session on Saturday, August 5, 1893, on which day there will be no divisional sessions.

The afternoons may be devoted to further sittings or to visits to the exposition or other points of engineering interest, as may be determined by the divisions.

It is expected that the evenings will be given up to receptions and social intercourse.

The Secretary of the Division will be glad to furnish any desired information upon application.

The following is a list of authors and papers for Division G, Marine and Naval Engineering and Naval Architecture, of the International Engineering Congress:

AUTHORS—FOREIGN.

Barnaby, Sir Nathaniel, K. C. B., Late Director of Naval Construction, British Navy.
The Best War Vessel.

Barnaby, S. W., with Thornycroft and Co.
Screw Propulsion.

Benbow, Henry, D. S. O., Chief Inspector of Machinery, British Navy.
Application of Forced Draft to Boiler Furnaces, its Effects in Causing Leaky Tubes, and the Remedies Therefor.

Biles, Prof. J. Harvard, Prof. Naval Architecture, University of Glasgow, Designer of the *City of New York* and *City of Paris*.
Fast Transatlantic Steamers.

Bona, Casimiro de, Inspector General of Engineers, Spanish Navy.
Rossin's Method of Graphical Integration Applied to Stability Calculations.
Bona's Elliptical and Circular Diagrams of Steam Distribution.

Busley, Carl, Professor in Imperial German Naval Academy, at Kiel, Germany.
A Marine Engineering Subject not yet stated.

Denny, Archibald, Partner in Wm. Denny and Bro.
Subject not yet stated.

Elgar, Francis, LL. D., Consulting Naval Architect, 113 Cannon St., London, England.
The present position of the Science of Naval Architecture.

Foley, Nelson, Manager Hawthorn Guppy Co., of Naples.
Review of the Rules for Boiler Construction of the Various Governments and Registration Societies.

Haack, Herr R., Late Technical Director of the Vulcan Shipbuilding Company, at Stettin, Germany.
The Trireme at the time of the Peloponnesian War.

Howden, James, Engineer, Designer of the Forced Draft System in use on the *City of New York* and *City of Paris*, and *Teutonic* and *Majestic*.
Forced Draft.
Resistance of Ships.

Liddell, Arthur R.
Practical Stability Information.

Mansel, Robert
Relation of Speed and Power in Steam Vessels.

Martell, Benj., Chief Surveyor of Lloyd's Register.
Naval Architecture in the United States.

Middendorf, Herr Fred. L., Technical Director of the Germanischer Lloyds.
On the Strength of Ships.

Migliardi, Signor G., Late Engineer Officer in Italian Navy; now of firm of Migliardi Bros.
Hydraulic Appliances in Connection with Modern Boiler Work.

Millar, W. J., Secretary, Institution of Engineers and Shipbuilders, in Scotland.
The Use of Oil at Sea.

Milton, J. T., Chief Engineer Surveyor of Lloyd's Register.
Comparison of Machinery for Naval and Mercantile Vessels.

Poli, Rodolfo, of firm of Poli Brothers, Chioggia, Italy.
The Coasting Sailing Vessels of the Adriatic Sea.

Richn, Prof. W., Professor at the Imperial Technical High School, Hannover, Germany.
The Resistance of Ships. Propelling Instruments.

Schlick, Consul O., Agent of Bureau Veritas, at Hamburg.
Vibration of Steamers, and his Apparatus (the Pallograph) for recording them.

Seaone, P. P., Cammandant of Engineers in the Department of Ferrol, Spanish Navy.
Diagram of Stability for any Draught and Stowage.

Seaton, A. E., Managing Director, Earle's Ship and Engine Building Company.
Multiple Expansion Engines, a Review of their History and their Probable Development.

Soliani, Colonel Nabor, Corpo del Genio Navale, Italian Navy.
The Use of Liquid Fuel on the Vessels of the Italian Navy.

Weir, Jas., of firm of G. & J. Weir, Glasgow.
Evaporators and Feed Water Heaters.

Ziese, Carl, Manager Schichau Works, which have built the fastest Torpedo Boats in the World.
Torpedo Boats.

Zimmerman, Rob't, Director of the Germannia Works, Kiel, Germany.
Ship's Scantlings.

AUTHORS—AMERICAN.

Ayres, S. L. P., Chief Engineer, U. S. Navy.
Conant, F. H , Assistant Engineer, U. S. Navy.
The Standardization of Steam-Engine Indicators and the Application of the Results in the Correction of Indicator Diagrams.

Beavor-Webb, J., Designer of the Yachts *Galatea* and *Genesta*.
Modern Steam and Sailing Yachts.

Chasmar, Jas. H., Chief Engineer, U. S. Navy.
The Various Iron Mixtures used in the Best Foundry Practice.

Cowles, William, President Cowles Engineering Co.
Speed Recorders for Ships.

Cramp, Edwin S., Superintending Engineer, Wm. Cramp and Sons.
Steel Castings for Machinery.

Denton, Jas. E., Professor in Stevens Institute.
Cylinder Condensation in Steam Engines, Historically Considered.

Dickie, Geo. W., Manager Union Iron Works, San Francisco, Cal.
Auxiliary Machinery on War Vessels.

Durand, Prof. W. F., Professor Marine Engineering and Naval Architecture, Cornell University.
Planning and Equipment of a Modern Ship and Engine-Building Plant.

Everest, Chas. M., Vice-President Vacuum Oil Company.
Oils and Lubrication.

Hill, Warren E., Vice-President Continental Iron Works.
The Strength of Welded Seams.

Hollis, Ira N., Past Assistant Engineer, U. S Navy, Assistant to Engineer-in-Chief.
Problems Confronting the Designer of Naval Machinery and the Success which has been Attained in their Solution.

Howard, Jas. E., Engineer of Tests, Watertown Arsenal.
Standard Form of Test Piece for Material Used in Connection with Marine Machinery.

Isherwood, B. F., Chief Engineer, U. S. Navy.
The Steam Jacket; its Genesis; its Principle of Action; and its Limitations.

McDougall, Alex., Inventor of the "Whaleback" system of hulls.
Whaleback Steamers.

Miller, Walter, Superintending Engineer, Globe Iron Works, Cleveland, Ohio.
The Steam Shipping of the Great Lakes.

Nixon, Lewis, Superintending Naval Architect, Wm. Cramp & Sons, Philadelphia.
The New Battle Ships and Cruisers of the U. S. Navy.

Rogers, Archibald.
Ice Yachts.

Smith, David, Chief Engineer, U. S. Navy.
A Standard Steam Engine Indicator and the Necessity therefor.

Stratton, E. Platt, Chief Engineer Surveyor to Record of American and Foreign Shipping.
The Government Inspection of Merchant Steamers, and the Influence thereon of the Rules of the Registration Societies.

Sweeney, John M.
Light Draught Steamers in use on the Western Rivers of the United States.

Taylor, Stevenson, Vice-President and General Manager A. & W. Fletcher Co.
Modern American Side-Wheel Steamers.

Towne, N. P., Chief Engineer, U. S. Navy; Consulting Engineer to the William Cramp & Sons Ship and Engine Building Company, Philadelphia, Pa.
Valves and Valve Gears for High Speed Engines.

Ward, Charles, Builder of the Tubulous Boilers of the U. S. S. *Monterey*.
Tubulous Boilers.

Weaver, W. D., Electrical Engineer; Formerly of Engineer Corps, U. S. Navy.
Speed and Revolution Recorder for Measured Mile Trials.

Willson, A. A., Superintending Engineer, Quintard Iron Works, New York.
Hydraulic Machinery for War Vessels.

Organization of Division G, International Engineering Congress, Division of Marine and Naval Engineering and Naval Architecture.

Chairman, **George W. Melville**, Engineer-in-Chief, U. S. Navy.

Secretary, **Walter M. McFarland**, Passed Assistant Engineer, U. S. Navy.

HONORARY PRESIDENTS.

UNITED STATES.

Copeland, C. W., Late Engineer-in-Chief, U. S. N., No. 24 Park Place, New York City.

Haswell, C. H., Late Engineer-in-Chief, U. S. N., No. 42 Broadway, New York City.

Isherwood, B. F., Chief Engineer, U. S. N., No. 111 E. 36th St., N. Y. City.

King, Jas. W., Chief Engineer, U. S. N., No. 3231 Powellton Ave., Phila.

Loring, Chas. H, Chief Engineer, U. S. N., 239 Clermont Ave., Brooklyn, N. Y.

Shock, W. H., Chief Engineer, U. S. N., No. 1404 15th St., Washington, D. C.

ENGLAND.

Barnaby, Sir N., K. C. B., Late Director of Naval Construction, R. N., Moray House, Belmont Hill, Lewisham, London, S. E.

Durston, A. J., Esq., Engineer-in-Chief, R. N., The Admiralty, Whitehall, London.

Reed, Sir E. J., K. C. B., Late Director of Naval Construction, R. N., Broadway Chambers, Westminster, London, S. W.

White, W. H., Esq., C. B., Director of Naval Construction, R. N., The Admiralty, Whitehall, London.

Wright, Sir James, K. C. B., Late Engineer-in-Chief, R. N., Dalcote, Arundel Road, Eastbourne, London.

FRANCE.

De Bussy, L., Inspecteur Général du Génie Maritime (en Retraite), 7 Rue de Jouy, Paris.

GERMANY.

Dietrich, Commodore Alfred, Chief Constructor, Imperial German Navy, Reichs Marine Amt., Berlin, Germany.

ITALY.

Brin, His Excellency Benedetto, Commodore Italian Navy, Minister of Foreign Affairs, Ministero Degli Affari Esteri, Rome.

Bozzone, Antenore, Commodore Italian Navy, Ministero della Marina, Rome.

Pucci, Guglielmo, Commodore Italian Navy, Ministero della Marina, Rome.

Racchia, His Excellency Admiral C. A., Minister of Marine, Ministero della Marina, Rome, Italy.

Sigismondi, Ippolito, Commodore Italian Navy, General Director of Naval Construction, Ministero della Marina, Rome, Italy.

JAPAN.

Sassou, Commodore T., Chief Constructor, I. J. N., Navy Department, Tokio, Japan.

SPAIN.

Bona y Garcia de Tejada, His Excellency Sr. Don Casimiro de, Inspector General de Ingenieros de la Armada, Ministerio de Marina, Madrid.

Bocas y Montero, Senor Don Dario, Engineer-in-Chief of Construction, Spanish Navy, Ministero de Marina, Madrid.

NORWAY.

Blom, Captain H. A., Chief Constructor, Norwegian Navy, Naval Department, Christiania, Norway.

Nielsen, Captain Alf, Director Naval Mechanical Establishments, Norwegian Navy, Christiania, Norway.

ADVISORY COUNCIL.

FOREIGN MEMBERS.

Barnaby, Sydney W., with Thornycroft & Co., The Hollies, Chiswick Mall, Chiswick, Middlesex, England.

Benbow, Henry, D. S. O., Chief Inspector of Machinery, R. N., Merton House, Ringmer, Sussex, England.

Biles, Prof. J. Harvard, Professor Naval Architecture, University of Glasgow, Glasgow, Scotland.

Blechynden, A., Superintending Engineer, Naval Construction and Armaments Co., Barrow in Furness, England.

Busley, Prof. Carl, Professor Marine Engineering, Imperial German Naval School, Kiel, Germany.

Corner, Jno. T., Fleet Engineer, R. N., Chief Engineer, Royal Dockyard, Portsmouth, England.

Cotterill, Prof. Jas. H., Professor App. Mech., Royal Naval College, Greenwich, England.

Denny, Archibald, Naval Architect, Wm. Denny & Bros., Dumbarton, Glasgow, Scotland.

Dredge, Jas., Editor Engineering, British Commissioner World's Columbian Exposition.

Dunn, Jas., Chief Constructor, R. N., The Admiralty, Whitehall, London, England.

Dunell, Geo. R., Mechanical Engineer, Staff of Engineering, No. 9, Grove Park Terrace, Chiswick, London, S. W., England.

Elgar, Francis, LL. D., Consulting Naval Architect, 113 Cannon St., London, England.

Foley, Nelson, Managing Director, Hawthorn Guppy Co., Naples, Italy.

Froude, R. E., Director, Admiralty Experimental Works, Haslar, England.

Gorris, Prof. J., Inspector, Technical Highschool, Charlottenberg (Berlin), Germany.

Haack, R., Naval Architect, No. 56 Ansbacherstrasse, Berlin, W., Germany.

Howden, James, Marine Engineer, Inventor of Howden's System of Forced Draft, No. 8 Scotland St., Glasgow, Scotland.

Kennedy, Prof. A. B. W., F. R. S., Consulting Engineer, No. 19 Little Queen St., Westminster, S. W., London.

Langner, Herr, Marine Engineer, Imp. German Navy, Principal Assistant to Director of Naval Construction, Reichs Marin Amt., Berlin, Germany.

Manuel, G. W., R. N. R., Superintending Engineer, P. & O. S. S. Co., 122 Leadenhall St., London, E. C., England.

Marshall, J. B., Chief Constructor, R. N., The Admiralty, Whitehall, London, England.

Martell, Benj., Chief Surveyor to Lloyd's Register, No. 2 White Lion Court, Cornhill, London, E. C., England.

Maw, W. H., Editor of Engineering, No. 35 Bedford St., Strand, London, England.

Middendorf, F. L., Director Germanischer Lloyd, No. 52 Markgrafen Strasse, Berlin, W., Germany.

Migliardi, Giovanni, Marine Engineer, Fratelli Migliardi and O. Vene, Savona, Italy.

Millar, W. J., Secretary Institution of Engineers and Shipbuilders in Scotland, 261 West George St., Glasgow, Scotland.

Milton, J. T., Chief Engineer Surveyor to Lloyd's Register, 2 White Lion Court, Cornhill, London, E. C., England.

Mudd, Thos., Superintending Engineer, Central Marine Engine Works, West Hartlepool, England.

Oldknow, R. C., Fleet Engineer, R. N., 139 Shooter's Hill Road, Blackheath, London, S. E., England.

Quick, George, Fleet Engineer, R. N. (retired), Glenbank, Bournemouth, England.

Richardson, Wigham, Naval Architect, Wigham Richardson & Co., Newcastle on Tyne, England.

Richn, Prof. W., Professor in Imperial Technical High School, Hannover, Germany.

Rudd, Chas., Staff Engineer, R. N., Controller's Department, The Admiralty, Whitehall, London, England.

Schlick, O., Agent Bureau Veritas, No. 7 Dammthorstrasse, Hamburg, Germany.

Seaton, A. E., Managing Director Earle's Ship and Engine Building Co., Hull, England.

Smith, W. E., Chief Constructor, R. N., Controller's Department, The Admiralty, Whitehall, London, England.

Soliani, Colonel Nabor, Corpo del Genio Navale della Marina Italiana, Ministero della Marina, Rome, Italy.

Traill, Thos. W., Fleet Engineer, R. N., M. Inst. C. E., Engineer Surveyor-in-Chief Board of Trade, Inspector of Chain Cable and Anchor Proving Establishments, No. 16 The Chase, Clapham Common, S. W., London, England.

Voss, Ernst, Marine Engineer, Blohm & Voss, Hamburg, Germany.

Williamson, J. C., Director of Dockyards, Royal Navy, The Admiralty, Whitehall, London.

Yarrow, A. F., Marine Engineer, Yarrow & Co., Isle of Dogs, Poplar, England.

Ziese, Carl, General Manager Schichau's Torpedo Boat Works, Elbing, Germany.

Zimmerman, R., Director Germania Works, Kiel, Germany.

AMERICAN MEMBERS.

Ayres, S. L. P., Chief Engineer, U. S. N., Navy Yard, New York.

Bailey, F. H., P. A. Engineer, U. S. N., in charge of Designs, Bureau of Steam Engineering, Navy Department.

Baird, Geo. W., Chief Engineer, U. S. N., U. S. S. *Dolphin*.

Baker, C. H., Chief Engineer, U. S. N., 1739 19th St., Washington, D. C.

Barber, F. M., Commander, U. S. N., Commanding U. S. S. *Monocacy*, Yokohama, Japan.

Beavor-Webb, J., Consulting Engineer and Naval Architect, No. 45 Broadway, New York City.

Boyd, Jas. T., Superintending Engineer, Manufacturing Department, Geo. F. Blake Manufacturing Company, East Cambridge, Mass.

Collins, Jno. W., Chief Engineer, U. S. R. M., Consulting Engineer, Revenue Marine Division, Navy Department, Washington, D. C.

Cooley, M. E., Professor Mechanical Engineering, University of Michigan, Ann Arbor, Michigan.

Cowles, Wm., President Cowles' Engineering Co., foot 44th St., Brooklyn, N. Y.

Cramp, Chas. H., President Wm. Cramp & Sons, Philadelphia, Pa.

Cramp, Edwin S., Superintending Engineer, Wm. Cramp & Sons, Philadelphia, Pa.

Davidson, Marshall T., Proprietor Davidson Pump Works, No. 43 Keap St., Brooklyn, N. Y.

Dickie, Geo. W., Manager Union Iron Works, San Francisco, Cal.

Dickinson, R. T., Superintending Naval Architect, Del. River Shipbuilding Co., Chester, Pa.

Doran, Jas. S, Superintending Engineer, International Navigation Co., No. 307 Walnut St., Philadelphia, Pa.

Durand, Wm. F., Professor Marine Engineering and Naval Architecture, Cornell University, Ithaca, N. Y.

Eckart, W. R., Consulting Engineer, No. 217 Sansome St., San Francisco, California.

Emery, Chas. E., Ph. D., Consulting Engineer, No. 915 Bennett Building, New York City.

Everest, Chas. M., Vice-President Vacuum Oil Co., Rochester, N. Y.

Farmer, Edward, Chief Engineer, U. S. N., Navy Yard, Portsmouth, N. H.

Fernald, F. L., Naval Constructor, U. S. N., Navy Yard, New York.

Fitch, H. W., Chief Engineer, U. S. N., Naval Academy, Annapolis, Md.

Gardner, Wm., Consulting Naval Architect, No. 1 Broadway, N. Y. City.

Grogan, F. W., Naval Architect, Architect Navy Department Exhibit, World's Columbian Exposition, Jackson Park, Chicago, Ill.

Henderson, Alex., Chief Engineer, U. S. N., Navy Yard, Boston, Mass.

Hill, Warren E., Vice-President Continental Iron Works, Brooklyn, N. Y.

Hollis, Ira N., P. A. Engineer, U. S. N., in charge of Designs, Bureau of Steam Engineering, Navy Department, Washington, D. C.

Holloway, J. F., Vice-President Worthington Pump Co., No. 145 Broadway, N. Y. City.

Hyde, Chas. E., Superintending Engineer Bath Iron Works, Bath, Maine.

Jewell, Theo. F., Commander, U. S. N., Inspector Ordnance, Navy Yard, Washington, D. C.

Kafer, John C., Superintending Engineer Morgan Iron Works, 9th street and East River, N. Y. City.

Kutz, Geo. F, Chief Engineer, U. S. N., Senior Inspector of Machinery, Union Iron Works, San Francisco, Cal.

Leavitt, E. D., Consulting Engineer, No. 2 Central Square, Cambridgeport, Mass.

Levy, Edw. L., Neafie and Levy, Philadelphia, Pa.

McElmell, Jackson, Chief Engineer, U. S. N., President Board of Examining Engineers, Philadelphia, Pa.

McMakin, Jos., Superintending Engineer Columbian Iron Works, Baltimore, Md.

Mallory, Geo. B., Consulting Engineer, No. 50 Broadway, N. Y. City.

Malster, W. T., President Columbian Iron Works, Baltimore, Md.

Mattice, A. M., Consulting Engineer, No. 2 Central Square, Cambridgeport, Mass.

Moore, Jno. W., Chief Engineer, U. S. N., Navy Yard, Mare Island, Cal.

Mosher, C. D., Naval Architect, No. 1 Broadway, N. Y. City.

Nixon, Lewis, Superintending Naval Architect, Wm. Cramp and Sons, Philadelphia, Pa.

Palmer, N. F., President Quintard Iron Works, New York City.

Pankhurst, J. F., Vice-President and General Manager Globe Iron Works, Cleveland, Ohio.

Perry, Jas. H., P. A. Engineer, U. S. N., Principal Assistant to Engineer-in-Chief, U. S. N., Bureau Steam Engineering, Navy Department, Washington, D. C.

Pusey, C. W., President Pusey & Jones Co., Wilmington, Delaware.

Quintard, Geo. W., Partner, Quintard Iron Works, New York City.

Raynal, Alfred, H., Superintending Engineer, S. L. Moore & Sons Co., Elizabethport, N. J.

Robinson, Lewis W., Chief Engineer, U. S. N., Chief of Department of Machinery, World's Columbian Exposition, Chicago, Ill.

Rowland, Thos. F., President Continental Iron Works, Brooklyn, N. Y.

Scott, Irving M., Vice-President and General Manager Union Iron Works, San Francisco, Cal.

See, Horace, Consulting Engineer, No. 1 Broadway, New York City.

Shaw, T. Jackson, Superintending Engineer, Harlan & Hollingsworth Co., Wilmington, Del.

Smith, David, Chief Engineer, U. S. N., Steel Inspection Board, Navy Department, Washington, D. C.

Spangler, H. W., Professor Mechanical Engineering, University of Pennsylvania, Philadelphia, Pa.

Stahl, A. W., Naval Constructor, U. S. N., Union Iron Works, San Francisco, Cal.

Taylor, Stevenson, Vice-President and General Manager, W. & A. Fletcher Co., Hoboken, N. J.

Thurston, R. H., Doc. Engineering, Director Sibley College, Cornell University, Ithaca, N. Y.

Towne, N. P., Chief Engineer, U. S. N., Consulting Engineer, Wm. Cramp & Sons, Philadelphia, Pa.

Van Buren, J. D., Consulting Engineer, Newburgh, N. Y.

Ward, Chas., Inventor and Builder of Ward's Tubulous Boiler, Charleston, Kanawha Co., West Virginia.

Waterhouse, W. E., Consulting Naval Architect, No. 50 State St., Boston, Mass.

Webster, H., Chief Engineer, U. S. N., Bureau Steam Engineering, Navy Department.

Wheeler, F. M., Mechanical Engineer, Secretary Geo. F. Blake Manufacturing Co., No. 95 Liberty St., New York City.

Willson, A. A., Superintending Engineer Quintard Iron Works, N. Y. City.

Worthington, Walter F, P. A. Engineer, U. S. N., Instructor in Marine Engineering, Naval Academy, Annapolis, Md.

STYLE No. 1.
The most modern instrument embodying the latest improvements. $85.00 net.

Edson's Pressure-Recording Gauge,

FOR

STEAM, WATER, GAS, AIR, OIL AND AMMONIA PRESSURE.

The only successful and recognized Standard Pressure Recorder, having been over 20 years in use, in all parts of the world

Indispensable where fuel economy and monitorial value is desired.

200 employed by National Transit Co.
81 employed by Illinois Steel Co.
67 employed by U. S. Government.
52 employed by British Government.
5 empl'd by Marshall, Field & Co., (Chicago.)

Also by the *Paris*, *New York*, *Friesland*, *Berlin*, *Columbia*, *Russia*, *Fürst Bismarck*, *Suevia*, *Normannia*, and other steamships.

Address for full particulars the sole proprietor and inventor,

JARVIS B. EDSON, PATENTEE,
87 Liberty St., New York, N. Y.

THE CONTINENTAL IRON WORKS

THOS. F. ROWLAND, President. WARREN E. HILL,
THOS. F. ROWLAND, JR., Treasurer. CHAS. H. CORBETT, Vice-Presidents.

WEST AND CALYER STREETS,

Near East 10th and 23d Sts. Ferries from New York.

BROOKLYN, N. Y.

SOLE MANUFACTURERS IN THE UNITED STATES OF

CORRUGATED FURNACES

FOR

MARINE AND LAND BOILERS.

In sizes from 28 to 60 inches in diameter, with flanged or plain ends.

The Kennedy Valve M'f'g Co.

52 CLIFF STREET, NEW YORK,

MANUFACTURERS OF

Brass and Iron Gate Valves, Indicator Valves,

Approved and in use in the U. S. Navy.

Fire Hydrants, Composition Castings, &c.

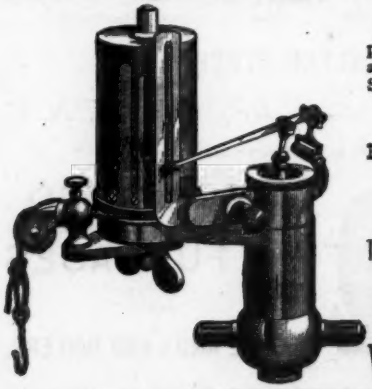
CONTRACTORS TO FOREIGN GOVERNMENTS.

Special Catalogue sent to Naval Engineers on application.

SCHÄFFER & BUDENBERG,

MANUFACTURERS OF THE

Straight Lyne Indicator.



Adapted for all speeds, unsurpassed for Simplicity, Reliability and Excellence of Workmanship. Sold at a Moderate Price.

TACHOMETERS.

Pressure Gauges for all Purposes, Engine Counters and Registers, Marine Clocks, Thermometers.

The Peerless and Manhattan Automatic Injectors, Reducing and Regulating Valves, &c.

WRITE FOR CATALOGUE.

WORKS: BROOKLYN, N. Y.

SALESROOMS:

No. 18 So. Canal Street, Chicago. No. 83 John Street, New York.

TRANSACTIONS OF THE DIVISION OF MARINE AND NAVAL ENGINEERING AND NAVAL ARCHITECTURE OF THE INTERNATIONAL ENGINEERING CONGRESS, HELD AT WORLD'S COLUMBIAN EXPOSITION, AUGUST, 1893.

Edited under the supervision of

COMMODORE GEORGE W. MELVILLE, Engineer-in-Chief, U. S. N.

Among the distinguished authors represented: Sir Nathaniel Barnaby, late Director of Naval Construction, British Navy; Henry Benbow, D. S. O., Chief Inspector of Machinery, British Navy; Prof. J. Harvard Biles, Professor Naval Architecture, University of Glasgow; Casimiro de Bona, Inspector-General of Engineers, Spanish Navy; Prof. Carl Busley, German Naval Academy, Kiel; S. W. Barnaby, Nelson Foley, Naples; James Howden, Prof. Andrew Jamieson, Benj. Martell, Chief Surveyor of Lloyd's Register; J. T. Milton, Chief Engineer-Surveyor, Lloyd's Register; Herr Fred. L. Middendorf, German Lloyd's; Signor G. Migliardi, late of Italian Navy; W. J. Millar, Rodolfo Poli, Chioggia, Italy; O. Schlick, Bureau Veritas, Hamburg; A. E. Seaton; Col. Nabor Soliani, Italian Navy; A. A. Willson, Carl Ziese, Schichau Works; J. Beavor-Webb, William Cowles, E. S. Cramp, Prof. J. E. Denton, George W. Dickie, Charles M. Everest, Stevenson Taylor, Warren E. Hill, Ira N. Hollis, James E. Howard, B. F. Isherwood, E. Platt Stratton, N. P. Towne, Chas. Ward and many others. 800 pages, 8vo, cloth, 200 illustrations and folding plates..... $10.00

To be published after the Session of the Congress by

JOHN WILEY & SONS,

53 East Tenth Street, New York.

R. BERESFORD,

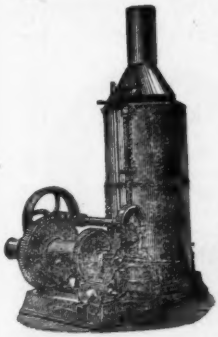
GENERAL PRINTER,

617 E STREET, N. W.,

CITY OF WASHINGTON.

The printing of Scientific Journals, Catalogues, Reports, Essays, Sermons, Etc., carefully and promptly executed.

COPELAND & BACON,



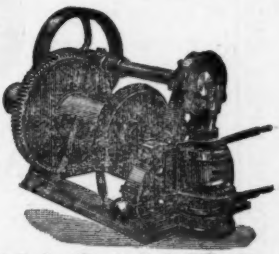
MANUFACTURERS OF

DECK HOISTS

STEAM

WINCHES

AND



HOISTING ENGINES.

NEW YORK: 85 LIBERTY.

CHICAGO: 62 S. CANAL ST.

SEND FOR CATALOGUE.

ESTABLISHED 1844.

A. SCHRADER & SON,

MANUFACTURERS OF IMPROVED

SUBMARINE ARMOR

AND

DIVING APPARATUS,

32 ROSE STREET, NEW YORK.

AN ABSOLUTELY PURE MINERAL COMPOUND.

FOR VALVES AND CYLINDERS.

LEAVES NO GREASE OR DEPOSIT IN BOILERS.

TRADE COLGAN LUBRICATING BALL. MARK.

COLGAN MANUFACTURING CO.,

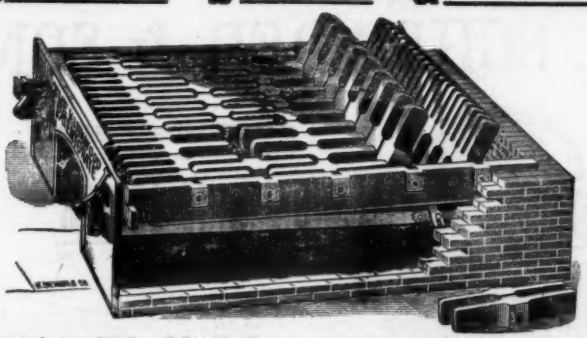
No. 234 West Street, Corner Beach,

NEW YORK.

Approved and recommended for use in U. S. Navy by the Experimental Board, Navy Yard, New York. Send for copy of report.

KIRKWOOD SHAKING GRATE-BAR

PATENTED.



PATENTED.

Approved by U. S. Navy Department for use on the New Cruisers of the U. S. Navy.

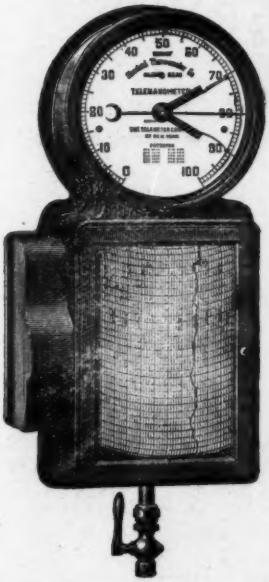
SEND FOR CATALOGUE.

KIRKWOOD GRATE-BAR CO.

185-189 Mercer St., NEW YORK.

Self-Contained

Electric Recording Pressure Gauge.



THIS gauge is for use where it is not desired to transmit the pressure to a distance.

It possesses merits not found in any mechanical recording gauge. The pressure spring is not hampered by a recording pen at the end of a long arm, which cannot fail to be a drag on the spring. It has a six-inch indicating dial like the ordinary commercial steam gauge, and the largest record sheet for the purpose ever introduced, having a daily time record twenty-four inches long and a pressure record five inches wide, revolving at the rate of one inch per hour, showing ten inches of the sheet, or the previous ten hours' record, clearly and distinctly. Each sheet has the time and divisions clearly lined and numbered, so that they can be clearly read several yards away. They are quickly and easily changed, and are convenient to be filed away or bound in a book six inches by twelve. Weekly records are of the same size as the daily. The clocks are eight-day, easily regulated and wound.

STANDARD THERMOMETER CO.,
Peabody, Mass.

NEW YORK: 418, 18 Courtlandt St.
BOSTON: 422 John Hancock Building

WORTHINGTON PUMPS

FOR

MARINE SERVICE

Horizontal Boiler-Feed Pumps

Vertical Boiler-Feed Pumps

Tank Pumps — Low Service Pumps

Bilge Pumps

Sanitary Pumps

Circulating Pumps — Ballast Pumps

Air and Circulating Pumps

Wrecking Pumps — Independent Condensers

HENRY R. WORTHINGTON

86 and 88 Liberty Street — NEW YORK

THE SAMUEL L. MOORE & SONS CO.

CRESCENT

SHIP YARD AND IRON WORKS

ELIZABETHPORT, N. J.

Engineers, Ship Builders, Machinists and Founders.

HEAVY CASTINGS A SPECIALTY.

Sole Makers of the Delamater Propeller Wheels.

XU

OFFICERS OF THE SOCIETY.

President:

Chief Engineer H. Webster, U. S. Navy.

Secretary and Treasurer:

Passed Assistant Engineer Walter M. McFarland, U. S. Navy.

Librarian:

Assistant Engineer W. W. White, U. S. Navy.

Council:

Chief Engineer H. Webster, U. S. Navy.
Passed Assistant Engineer Walter M. McFarland, U. S. Navy.
Passed Assistant Engineer Frank H. Bailey, U. S. Navy.
Passed Assistant Engineer Ira N. Hollis, U. S. Navy.
Assistant Engineer W. W. White, U. S. Navy.

The subscription price of the Journal is $5.00 per annum, payable in advance.

Make cheques, drafts and postal orders payable to

AMERICAN SOCIETY OF NAVAL ENGINEERS,

except Naval Paymasters' drafts, which should be made payable to the officer who remits and *endorsed* to the Secretary-Treasurer by name, *without any title.*

Make postal orders payable at Station C, Washington, D. C.

Advertising rates will be furnished on application.

It is earnestly requested that prompt information be given of changes in address, or of failure to receive the Journal.

Address all communications for the Society to

AMERICAN SOCIETY OF NAVAL ENGINEERS,
Navy Department, Washington, D. C.

XU